

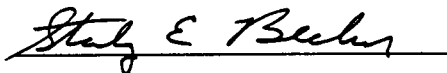
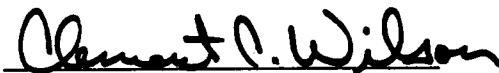
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Frank H. Speckhart,
Major Professor

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**THE DESIGN AND PROTOTYPING OF A LOW-COST
MOTORCYCLE CARGO TRAILER**

A Thesis
Presented for the
Master of Science
Degree
The University of Tennessee, Knoxville

Olakunle Harrison

December 1989

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ABSTRACT

In developing countries, the motorcycle is a major means of transportation among low income groups. Since for most of the people who ride motorcycles in these countries it is also the sole means of transportation, a motorcycle trailer would serve in hauling cargo safely and efficiently. The trailer should weigh about 100 pounds and have a payload of 200 pounds. Because many of the roads are substandard, the trailer must be able to traverse bad roads with minimum adverse effects on the motorcycle and its rider. Of necessity, the design must be easy to assemble, the cost of the trailer a minimum, and the manufacturing processes readily available in developing countries.

A low cost trailer of original design was developed, built and evaluated. The design possesses characteristics that reduce or eliminate the adverse effects of variability in the manufacturing process. Where feasible off-the-shelf parts and stock dimensions were used to achieve a unit cost of 136 dollars.

A trailing arm configuration sprung independently by rubber torsion springs was selected for the chassis. The chassis is designed to produce a lower center of gravity than available in most trailers in its category. The design allows for flexibility of application by accomodating different body styles on a flatbed base. A unique coupler that allows a large relative rotation between the motorcycle and trailer was also designed.

The evaluation involved tests and observations of trailer swing stability, steady-state turn stability and brake-in-turn behavior. The trailer's response to high shock loads was observed and recommendations of tire pressure in service determined.

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CHAPTER 1

INTRODUCTION

1.1 Background

In developing countries, the motorcycle features prominently in transportation. It is a very common means of transportation within the low income groups. Presently, in most cases, cargo is hauled by attaching it to the luggage carrier behind the seat. Cargo volume is thus limited to the small amount of space available. This limits the utility of the motorcycle considerably and can also present a safety hazard. By stacking loads up high on the rear of the motorcycle, the center of gravity of the system (motorcycle, rider and load) is elevated and the moment of inertia about the longitudinal axis of the motorcycle is increased. This setup can greatly increase the potential for accidents. A trailer pulled behind the motorcycle would serve to increase its utility without unduly jeopardizing the rider safety.

The motorcycle trailer is not uncommon on U.S. roads. Usually, they are found behind large touring bikes with about 1000 cc engine displacements. The high displacement of the engines corresponds to gross vehicle weights substantially higher than those of motorcycles in developing countries. These motorcycles are usually in the 250 cc displacement category. Thus the weight of the trailer becomes very important when designed for smaller motorcycles. For the combination of motorcycle and trailer to be safe, the trailer must be of sufficiently low gross weight.

Also, many of the roads in developing countries have surface irregularities that can increase the risk of accidents for a motorcycle pulling a trailer. Traversing these roads with a trailer requires that adverse effects of poor road surfaces be minimized. The trailer

suspension should not respond to bumps in such a way that excessive excursions of the trailer results.

Since this product is aimed primarily at low income consumers, it is important that the associated costs be low enough to be affordable. The manufacturing processes involved, capital equipment needed, labor and parts costs must all serve to produce a low cost, safe trailer. It should be durable and have low maintenance requirements.

1.2 Design Goals

The primary thrust of this project is the design of a low cost, high quality motorcycle trailer. The trailer should be approximately 3 feet wide by 3 1/2 feet long and have a payload of 200 pounds. The goal for the unladen weight is 100 pounds and the cost of parts and labor should be a maximum of 100 dollars.

Elements of design-for-manufacture are to be used to achieve the low-cost objective. Furthermore, the design should use as many stock parts and dimensions as possible. The trailer must be durable and withstand normal abuse by the consumer.

The trailer must be safe to tow and not present an undue hazard to the rider. In trailer dynamics, the four most important parameters affecting trailer performance are braking performance, trailer swing stability, steady-state turn stability and brake-in-turn response. Standards have been developed for automotive trailers by Klein and Szostak [1]* using these as performance criteria. Although the motorcycle differs considerably from the automobile, an attempt should be made to incorporate those aspects of the standards that are relevant to a motorcycle/trailer combination. The wheel track, tongue length, center of gravity and suspension should ensure a safe motorcycle/trailer combination.

* Numbers in [] refer to items in the list of references

1.3 Scope

This thesis focuses on the design and prototype of the motorcycle trailer. This includes engineering analyses of components, manufacturability and an evaluation of the prototype. It presents an overview of pertinent considerations in trailer design and performance, and examines typical configurations for this category of trailers. This project also includes the design of a swivel coupler that allows the motorcycle to rotate up to 45 degrees from the vertical with the trailer attached. The evaluation of the trailer involves observations of its response to various loading conditions over different terrains, trailer swing, braking and steering.

CHAPTER 2

CONSIDERATIONS IN TRAILER DESIGN

The two areas of concern in trailer performance are braking and handling. These have a direct impact on the safety of the motorcycle/trailer combination. Currently there are no standards for motorcycle trailers. Standards do exist for automotive type trailers. These standards address the safety of combination vehicles and were obtained primarily from empirical data from extensive performance tests. They focus on straight line braking, trailer swing, tow vehicle turn stability and braking-in-a-turn performance. Also of concern in trailer design is the type of suspension system used. Because it determines factors such as center of gravity, roll axis and wheel kinematics, it significantly affects the handling characteristics of the motorcycle/trailer combination.

2.1 Braking Performance

Straight line braking performance is important because of the increased stopping distance caused by the addition of the trailer mass. Under ideal conditions the deceleration of the combination vehicle is given in [1] as

$$a_{cv} = \frac{a_v W_v + a_t W_t}{W_v + W_t} \quad (2.1)$$

where

a_{cv} = Combination vehicle deceleration, g

a_v = Tow vehicle deceleration capability, g

a_t = Trailer-alone deceleration capability, g

W_v = Tow vehicle weight, lb

W_t = Total trailer weight, lb

For an unbraked trailer, a_t is zero. The tow vehicle is responsible for all braking. Thus the added weight of the trailer is a limiting factor. If the weight approaches that of the tow vehicle, there may not be sufficient braking capability for the combined vehicle. It is important that the total weight not exceed the vehicle manufacturer's gross vehicle rating. Furthermore, for cars and motorcycles alike, the location of the hitch plays an important role in braking performance. A hitch load that reduces the load on the front wheel(s) will decrease braking performance, since front brakes usually contribute upwards of 70 percent of the total braking forces. For this reason, automobile trailers that have load leveling devices are known to produce slightly improved stopping distances for the combination vehicle [1]. This is due to the larger vertical load carried by the front axle.

2.2 Trailer Swing Stability

Trailer damping is a measure of the rate at which the amplitude of oscillation of the trailer decreases. This oscillation takes place about a vertical axis through the hitch point. The oscillation is akin to the swing of a pendulum with its center at the hitch point. For a high tow vehicle weight to trailer weight ratio, this oscillation does not pose a threat to safety. However, the stability of a combination vehicle can be compromised when the weight of the trailer approaches the tow vehicle's weight. Damping ratio is the quantity used to measure the decrease in oscillation and should be positive for a stable combination vehicle.

Consider the case of a trailer attached to an immovable hitch. This is represented by the simplified model shown in Figure 2.1. The external torques acting on the system are the force arising from the tire lateral stiffness and a damping force at the tires. The magnitude of the damping force will depend on the amount of friction between the tires and the road surface, the weight directly on the tires and the trailer's forward velocity.

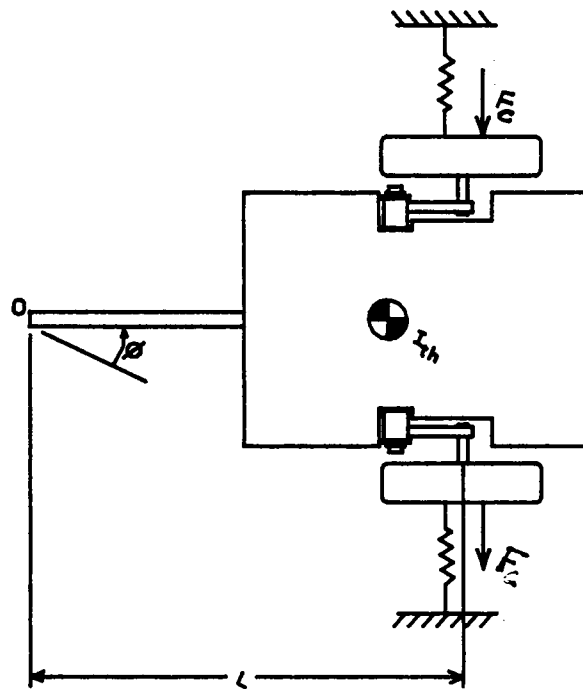


Figure 2.1. Dynamic model of a trailer.

The damping ratio ζ for this second-order system may be evaluated, [1], by

$$\zeta = \frac{\sqrt{L^3 Y}}{U \sqrt{2 I_{th}}} \quad (2.2)$$

where

I_{th} = trailer moment of Inertia, lb-s²-ft

L = hitch-to-axle distance, ft

Y = tire lateral stiffness, lb/rad

U = forward speed, ft/s

It can be seen from equation 2.2 that the most influential parameter governing swing is the hitch-to-axle distance L . The longer the hitch-to-axle distance, the higher the damping, and the slower the combination vehicle, the better. The trailer manufacturer has control over the hitch-to-axle distance and strives to make it as large as practical. Furthermore, from [1], stiffer tires and smaller moment of inertia about the hitch point yield increased damping. From experiments on automobile trailers, numbers of significance have been attached to damping ratios, with 0.15 at 55 miles per hour being the suggested criterion [1]. The suggested performance standards for trailer swing takes into account factors such as yaw motion, hitch loads, tire cornering stiffness, steering free play and more. Many of these factors are quite different in motorcycle dynamic characteristics, so that the application of these numbers to the motorcycle/trailer combination is neither straightforward nor necessarily valid. However, some of the parameters over which the manufacturer has control are to some extent independent of the tow vehicle. These include hitch-to-axle distance, the wheel track, and the hitch load. The influence of these parameters depend on the size and weight of the trailer as well as the type of suspension used.

2.3 Turn Stability

Steady-state turn stability and braking-in-a-turn performance are two other control and dynamic stability measures that are related to understeer and oversteer in the tow vehicle. The tendency of the combination vehicle to jackknife is exhibited in these stability criteria. In this respect, the dynamic characteristics of the motorcycle are markedly different

from that of the automobile. Thus, suitable tests should be made to determine if there is a marked degradation in the handling resulting from the addition of a trailer.

2.4 Suspension Principles

The suspension design also features prominently in the response of the trailer. The suspension requirements for a trailer are: freedom from unfavorable steering effects, reduction of strain on cargo and vehicle components, reduction of body accelerations and good cornering ability. The suspension must be capable of meeting its requirements, to some extent, on a variety of road surfaces. Smooth roads produce impact loads of 20 percent of the trailer weight, while rough roads produce up to 100 percent of the weight. Very rough roads can produce impact loads of 200 percent or more [2]. For a given road surface, the magnitude of the shock transmitted to the axle is influenced by tire diameter, tire contact patch size, tire spring rate and the weight of the wheel and associated suspension components. Thus a balance between high shock absorber forces and soft springing should be sought.

The wheel encounters two types of shock loads: one due to a bump and the other due to a pothole. The magnitude of a shock load due to a bump depends on the geometry of the bump as well as the speed of the vehicle. The intensity of the shock due to a pothole depends on the geometry of the pothole, the unsprung mass and the stiffness of the springs. Thus for improved response to potholes, low unsprung masses and soft springing is desirable.

Shock absorbers are used in suspension systems to reduce the number of oscillations of the sprung mass when the wheel encounters road surface irregularities as well as to reduce the amplitude of oscillation when the sprung mass is excited at frequencies at or close to its resonant frequency. The excitation at resonant frequency is caused by two

different phenomena: travelling over a washboard surface at a speed that coincides with the natural frequency of the sprung mass, and an unbalance in the wheel. Shock absorbers also reduce accelerations of the body and keep the wheels in contact with the road surface.

For an automobile the frequency of oscillation of the sprung mass is typically in the 1 to 3 hz range while that of the unsprung mass - the tire, wheel and suspension links - is generally about 15 hz [2]. The latter is called the wheel hop frequency. The motion of the carriage unit is to a large extent independent of wheel hop. To minimize wheel hop, the unsprung masses must be kept to a minimum. High shock absorber forces and soft tires also reduce wheel hop.

In many trailer suspension systems, there are no shock absorbers. Rather, the damping that is inherent in the tires and the friction between multileaf leaf springs reduce wheel oscillations. While softer tires reduce the shock loads transmitted to the carriage unit, they can adversely affect the cornering ability due to increased sideslip angles. This reduces the tire's ability to sustain higher lateral forces than it otherwise would.

Springs absorb forces exerted on the wheels by the road surface. The force transmitted to the chassis is the same as that causing the spring to deflect. Thus, the chassis accelerates, but at a much slower rate than a suspension without springs. Subsequent vibrations of the body are reduced by damping. Soft springing is used to reduce the shock transmitted to the carriage unit. If, however, the spring stiffness is too low, the lateral stability of the vehicle will be compromised. This is due to excessive body roll that will occur.

Several types of springs are used in small trailers. These include leaf springs, coil springs, rubber springs and torsion springs. The primary considerations in the type of spring used are installation costs, weight and relative energy storage capability.

2.5 Suspension Systems

In the design of a trailer suspension, there are many factors to be considered. These include the following:

1. Load-carrying capability
2. Kinematic considerations - wheel travel, camber, toe, etc.
3. Dynamic characteristics - lateral stability, wheel hop, body acceleration, steering effects, resonant frequencies
4. Suspension overload
5. Wheel alignment
6. Reliability and maintainability
7. Manufacturing costs

The optimum design parameters for a trailer suspension system are [3]:

- freedom from adverse steering effects from road surface irregularities and lateral forces,
- a slightly negative camber during cornering, along with minimum camber change,
- track changes as small as possible to provide lateral stability and minimal tire wear, and
- reduced shock loads transmitted to vehicle chassis and cargo.

Trailer suspensions can be divided into two categories: dependent and independent suspensions. In independent systems, the motion of one wheel does not affect the other. In the dependent system, the wheels interact with each other. The choice of suspension type will depend on a suitable compromise between the parameters listed above.

The dependent suspension system is one in which a solid beam or axle connects two wheels. It is used on most trucks and buses that must carry great loads. They are used

because such large vehicles have shallow frames and, hence, are not very stiff torsionally [2]. The two wheels are rigidly connected so that they lie parallel to each other and remain so for any given motion. The disadvantages of the dependent suspension system include:

1. Dynamic loads on one wheel affect the other thereby causing axle tramp.
2. Inadvertent steering effects when traversing rough terrain.
3. Tendency of the axle to rotate about a vertical axis on rough roads.

Its advantages are :

1. Simpler and more economical construction and manufacture.
2. No changes in track, toe and camber resulting in reduced tire wear.

The independent suspension system allows the designer better control of the parameters that yield optimum performance. Its advantages include.

1. The motion of one wheel does not affect the other.
2. Lower space requirements.
3. Lower unsprung mass.
4. Can be designed to produce a lower vehicle center of gravity without an increase in space requirements.
5. Lower roll center more easily obtained.

The most important disadvantage of the independent suspension is higher costs due to the increased complexity of the system. In most designs, it invariably involves more parts which lead to higher costs.

The suspension geometry should strive towards achieving optimum suspension parameters within cost limitations. Ideally, the camber of the wheels - inclination from the vertical axis- should start at zero in the unloaded position and become slightly negative as the loading increases [3]. This configuration improves the lateral stability of the vehicle.

There should be no track changes as this seriously affects wheel adhesion especially on wet surfaces. Also, track changes increase tire wear.

The roll axis of a vehicle is a line on a vertical plane containing the longitudinal axis about which the body will rotate when subjected to a lateral force [4]. The location of the roll axis is determined by the suspension geometry. Consider the rear view of the solid axle suspension system shown in Figure 2.2. When a lateral force F acts on the vehicle body, it will cause a couple Fh about point O . This point is the one about which the body will rotate since one spring will be compressed the same amount as the other is extended. Point O thus lies along the roll axis and is the roll center. With independent suspension, Figure 2.3, the same lateral force F causes the body to tilt as shown. Here, the roll axis is much lower since the body tilts about its instantaneous center O . For a vehicle with the roll axis at or near ground level, the couple required to overturn it will be greater than that of a vehicle

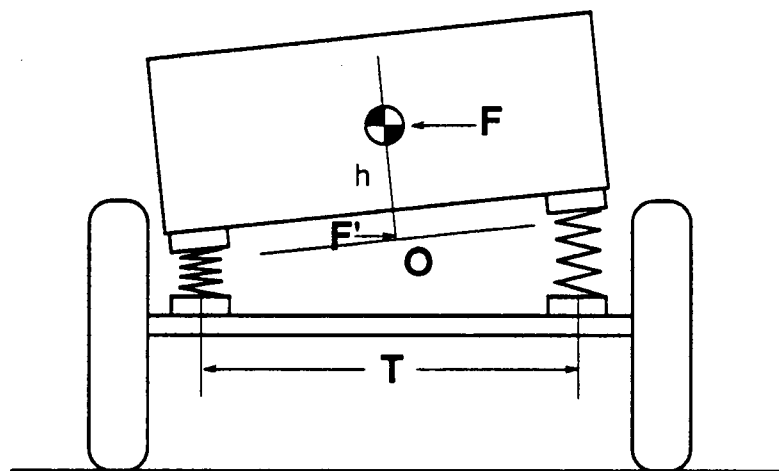


Figure 2.2. Schematic of a solid axle suspension system.

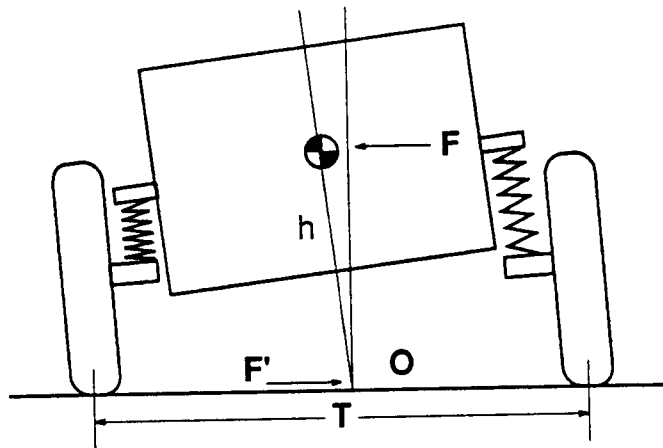


Figure 2.3. Schematic of an independent suspension system.

with a higher roll center [4]. But, the tilt of the vehicle with independent suspension will be greater than one without. This, however, is offset by the fact that the effective spring base T , is much greater for independent suspension. For this type of arrangement, the effective spring base is the wheel track, while for a rigid axle, it is the center distance between the two springs. The angle of tilt is approximately proportional to $2Fh/T^2$ [4]. Thus the influence of the effective spring base is greater than the effect of the moment arm h of the overturning couple.

The roll center is found by extending the axis of the suspension links to a common point p as shown in Figure 2.4. A line is then drawn from point p to the center of the tire contact patch. Since symmetry exists about the vehicle vertical centerline, the point at which the line from p to the contact patch intersects the vertical center line is the instantaneous center, point O . This is the roll center of the vehicle. For a rigid axle with leaf springs, the

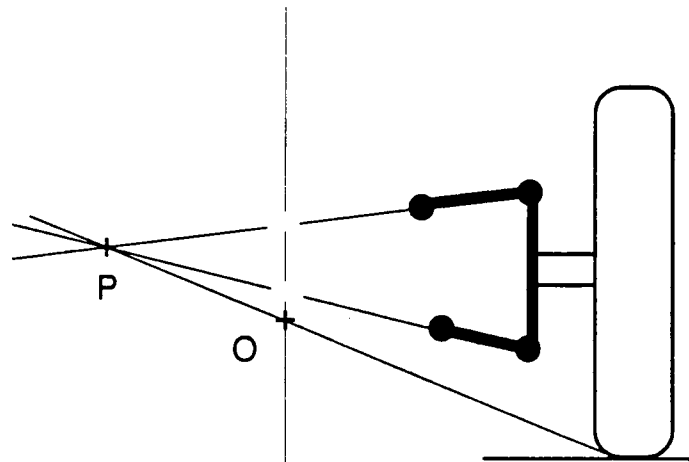


Figure 2.4. Locating the roll axis of a suspension system.

roll center lies at the spring eyes above the axle [4]. For a trailing arm arrangement, the center of rotation p for each wheel lies at an infinite distance from each wheel. A line drawn from point p through the tire contact patch intersects the vertical centerline at ground level, Figure 2.5. Hence the roll center for the trailing arm arrangement is at ground level.

2.6 Axle Location

The hitch-to-axle distance and the location of the axle along the trailer are very important factors in the stability of the trailer [1]. It is desirable to locate the trailer axle such that a downward vertical force is exerted on the hitch. If the hitch load is in the opposite direction it effectively reduces the vertical load on the vehicle's rear axle. This could result

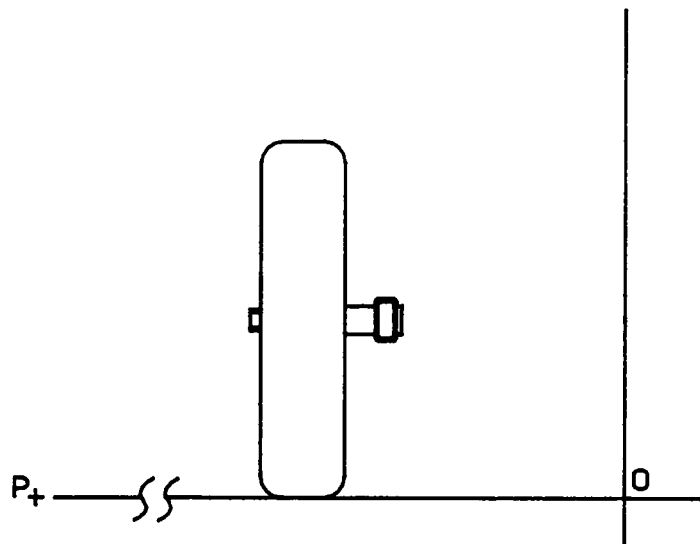


Figure 2.5. Roll center of trailing arm suspension.

in an unstable trailer. The manufacturer is limited to a practical hitch-to-axle distance and has no direct control over the load distribution. A hitch load of 10 percent is recommended for automobile trailers [1]. The axle is located such that this criterion is satisfied for the unloaded trailer. The 10 percent hitch load value is used in this design.

CHAPTER 3

DESIGN METHODOLOGY

3.1 Survey of Existing Trailer Configurations

By far the most common trailer design is the solid axle suspension system that uses semi-elliptic leaf springs. This is shown in Figure 3.1. The axle is mounted on the leaf springs which in turn are attached to the frame that carries the trailer body. The springs are mounted either above or below the axle beam depending on the desired ground clearance. U-bolts are used for this connection. Shock absorbers are not used. Standard spindles are welded to each end of the tubular axle.

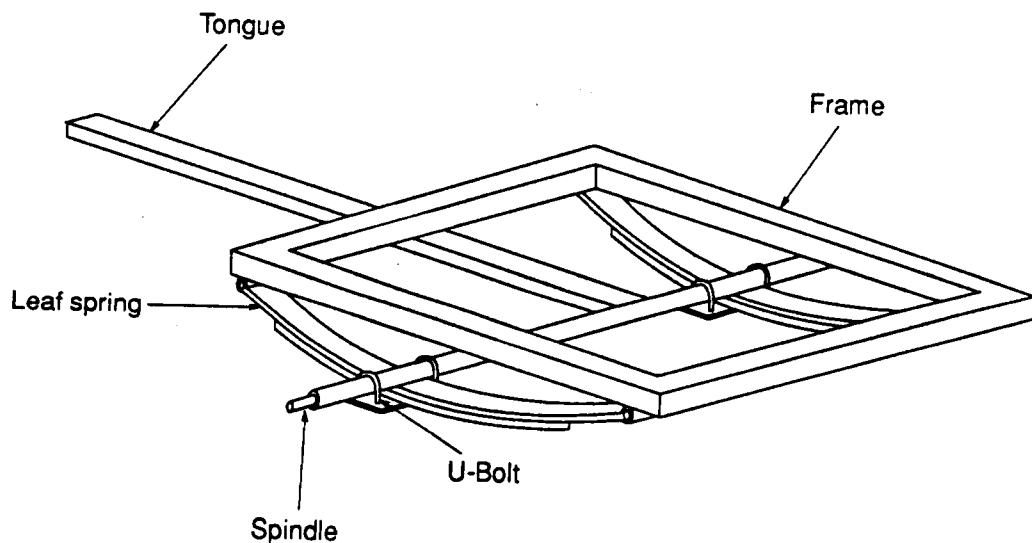


Figure 3.1. Conventional trailer chassis.

The straight trailer tongue is attached to the middle of the axle and frame, and is coupled to a hitch ball mounted on the motorcycle. There appears to be no standard hitch-to-axle distance for the different trailers surveyed. This distance ranges anywhere from 36 inches to 72 inches. In order to avoid increasing the unsprung masses, the hitch is connected to sprung members on the motorcycle, typically the frame and/or rear passenger foot pegs. The motorcycle rear axle is not used for this purpose. The hitch ball is located some distance behind the motorcycle rear wheel at a distance of about 13 inches from ground level. The hitch ball is supported by a truss member.

The most common wheels are designated 480-12 which, with tires mounted, have a diameter of approximately 16 inches. The wheels run on tapered roller bearing hubs that are attached to the spindles.

Either a triangular frame or a rectangular frame is used for most trailers. The frame members are usually of square cross-section and are either welded or bolted together to form the frame. The trailer body, usually fiberglass reinforced plastic, is then mounted on the frame. Average overall trailer dimensions are 36 inches width, 62 inches length and 30 inches height from ground level. The trailer unladen weight ranges from 125 pounds to 200 pounds.

3.2 Critique of Conventional Trailer Designs

The solid axle suspension that uses semi-elliptic leaf springs represents the least cost approach to the manufacture of a trailer. It results in very few parts and relatively few manufacturing operations. Aside from the tires, wheels, hubs and spindle (which every trailer uses) the required chassis parts are leaf springs, U-bolts, mounting plates and frame and tongue members.

Leaf springs are commonly used because of the following advantages:

1. Ability to absorb forces in three dimensions, thus locating axle.
2. Nonlinear load-deflection characteristic.
3. Inexpensive.

The disadvantages of leaf spring include

1. Low load carrying capability per unit weight.
2. Wear caused by sliding of leaves relative to each other.
3. Insufficient damping.
4. Unfavorable steering effects when travelling over rough terrain.

The suspension of the trailer is a very important factor in the stability of the motorcycle/trailer combination. For the large touring motorcycle, the addition of a 300 pound trailer may not pose a serious threat to rider safety. The smaller motorcycle, however, will benefit from a trailer suspension designed towards achieving optimum towing parameters. The suspension design determines factors such as the location of the center of gravity, amount of wheel hop and the roll center. The suspension also influences performance criteria such as swing stability, braking-while-turning, and pulse steering.

Perhaps the most voiced objection by consumers is about the trailer that bounces excessively over irregular road surfaces. The oscillations of the body may simply be more of a nuisance on relatively smooth roads. For many roads in developing countries, the potholes and bumps can be substantial enough to cause the trailer to develop motions out of line with the motorcycle. This problem is especially of importance when irregularities are encountered while turning. If, for example, a fully loaded trailer sidesteps excessively due to a bump on the road when cornering, the effect can be catastrophic to the combination vehicle.

The leaf spring most frequently used in motorcycle trailers measures 24 inches in length, is 1.5 inches wide and has 3/16 inch thick leaves. Two leaves are used. The deflection d of the semi-elliptic leaf spring is [3]

$$d = \frac{(2 + \frac{n}{n'}) E n b h^3}{6 L^3}$$

where

n - number of leaves

n' - number of leaves at end of spring

b - spring width

h - leaf thickness

L - distance from spring eye to center of spring

E - modulus of elasticity

An estimate of the spring rate shows a typical stiffness of 143 lbs/in. For both springs in parallel, then, the effective spring rate is 286 lbs/in. The springs have a rated capacity of 510 pounds each. On virtually all trailers using leaf springs, the absence of shock absorbers is obvious. For a trailer with an unladen weight of 125 pounds, it is easily seen that the springs will not deflect much when the trailer is empty or lightly loaded. With no shock absorbers present, the only significant damping is provided by the tires. Since the tires cannot have pressures too low on account of the load to be hauled, bouncing of the trailer is imminent. Furthermore, the stiff springs will result in the transmission of high forces to the chassis and body leading to increased wear and tear over a long period.

Using leaf springs also results in a center of gravity that is unnecessarily high from ground level. Since the motorcycle trailer is relatively small, it does not need a very large ground clearance. The center of gravity should be as low as practicable to improve lateral

stability. Also, the use of leaf springs results in a high roll center since the roll center lies at the point of attachment of the springs to the chassis.

The location of the hitch is another aspect of the conventional trailer design that deserves attention. It is commonly located behind the motorcycle rear wheel at a distance of about 13 inches from ground level. This location has two drawbacks. The first problem is that the hitch load reduces the weight on the front wheel of the motorcycle due to the applied moment about the motorcycle center of gravity. This weight reduction can adversely affect the steering and braking of the motorcycle. Secondly, the truss-like hitch arrangement protrudes behind the motorcycle and may not be appealing or safe when the trailer is not attached to the motorcycle. Most of the motorcycles for which this trailer is being designed do not have saddle bags and body accessories that make the hitch less obvious or intrusive with the trailer detached.

An examination of a conventional automotive-type coupler showed that the relative rotation between the motorcycle and trailer is limited to about 12 degrees. This represents the maximum angle that the rider can lean in order to negotiate a turn without being constrained by the trailer. Also, in the likely event that the motorcycle tips over when parked, the coupler will be subjected to torsional stresses. If this happens too often, the coupler may fail in service.

3.3 Design Approach

The suspension system is a very important aspect of trailer design. Once determined, all other parameters such as hitch-to-axle distance and the longitudinal location of the center of gravity can be varied to achieve the desired effects. An independent suspension system is the better way of obtaining near-optimum parameters of a suspension system. In light of the need to design a safe trailer that can be towed by a small motorcycle without

unnecessarily increasing manufacturing costs, an independent suspension system was chosen. Even though it might not be the least expensive suspension system, it will be better able to meet the requirements that produce a high quality product. The design goal is to achieve near-optimum parameters without an excessive increase in the cost over the dependent suspension system that is the most common configuration.

One of the least expensive ways of achieving independent suspension for a trailer is by the use of a trailing arm configuration. As shown in chapter 2, this type of design puts the roll center of the trailer at ground level. The force required to overturn such a trailer will be greater than that of the semi-elliptic spring, dependent suspension. The center of gravity of the trailer is considerably lower in this arrangement. For a trailer sprung by leaf springs, the trailer bed is at least 12 inches from ground level. This assumes the spring has a 4 inch deflection and is mounted on an axle which is 8 inches from the ground at the wheel center. For the suspension designed here, the trailer bed is 8.5 inches from ground level and can be varied as desired during manufacture without much of a problem. The trailing arm suspension generally produces zero track and camber changes.

Several types of springs were considered for the suspension. These include helical springs, torsion bar springs and rubber springs. Coil springs have better load carrying capabilities than leaf springs (a 3:1 ratio in their favor,) but were rejected because, for a trailer so sprung, several other components are needed to locate the axle and wheel. This leads to an increase in the number of parts needed to suspend such a system. They also have no significant damping qualities and would thus require some additional means of damping.

Torsion bar springs cannot be made simple enough for a low cost trailer. This is because of unacceptable stresses resulting from the desired deflection of about 3 inches. Consider the torsion bar arrangement shown in Figure 3.2. For an applied force P at the

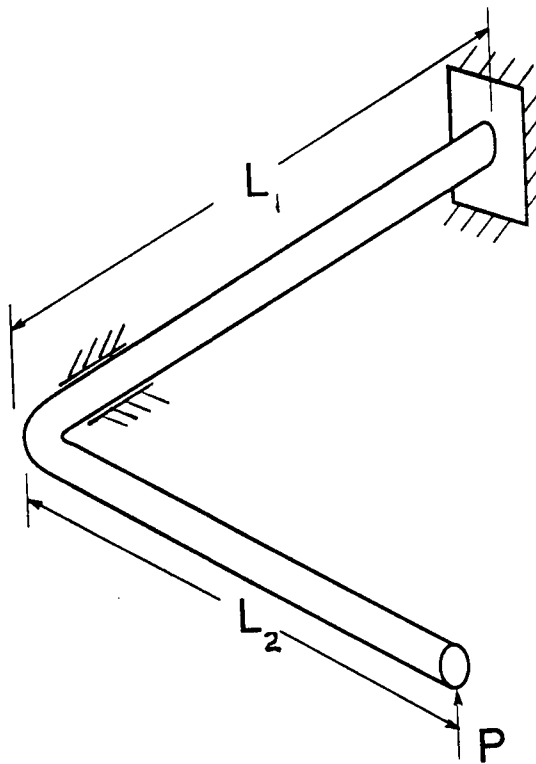


Figure 3.2. Torsion bar loading.

point shown, the deflection of point A is the sum of deflections due to bending of member AB and the torsion in rod BC. The resulting expression for the total deflection is

$$d = \frac{PL_2^3}{3EI} + \frac{TL_1L_2}{GJ}$$

For $L_1 = 8$ inches, $L_2 = 8$ inches and an overload P of 250 pounds, the expression produces a required torsion bar diameter of 0.54 inches. For a bar of this diameter subjected to a moment of 2000 in-lb, the resulting shear stress is 129,136 psi. A bending moment of 2000 in-lb produces a normal stress of 129,374 psi. Thus it can be seen that these stresses are more than double the yield stress for steel. The torsion bar cannot be designed for the desired deflection in the available space. Because of this problem, torsion bars are sometimes bundled together to achieve large deflections. This approach, however, results

in a complicated arrangement and is deemed too expensive for this application. Torsion springs also require special manufacturing operations, such as shot-peening, for surface protection. Furthermore, they do not have any self-damping.

Torsional rubber springs were chosen for the spring elements because of their characteristics. Figure 3.3 shows the rubber spring arrangement. Rubber springs offer a nonlinear, progressive spring rate. The torsional rubber spring is initially loaded in shear. As the loading progresses, the compressive mode in which rubber is stiffer, comes into play. This allows the unloaded trailer to ride without excessive bounce over rough roads. The loaded trailer benefits from the increased spring rate. Rubber springs are compact and have a much higher energy absorbing capability per weight ratio than leaf springs. Typically, they offer between a 7:1 to 13:1 ratio over leaf springs [5]. Rubber springs possess a considerable amount of self-damping. This characteristic makes them cost effective in the design since separate shock absorbers can be avoided. The design of the suspension also takes advantage of the high stiffness of rubber in compression. By having a moment applied in the radial direction of the spring, a small amount of wheel camber is introduced. This is considered beneficial for lateral stability. A slightly negative camber during cornering improves the lateral stability of the trailer. Rubber springs also have the advantage of being maintenance-free. They do however tend to take a set over very long periods of time. Discussions with an engineer representing the Lord Corporation- a manufacturer of rubber springs - led to the conclusion that this is not a significant problem.

The trailer tongue design involves an elevated shape, Figure 3.4, that allows the trailer to be hitched to the rear luggage rack common to most small motorcycles in developing nations. With this arrangement, the hitch load is applied at a point much closer to the motorcycle center of gravity than afforded by a hitch located behind the rear wheel. A

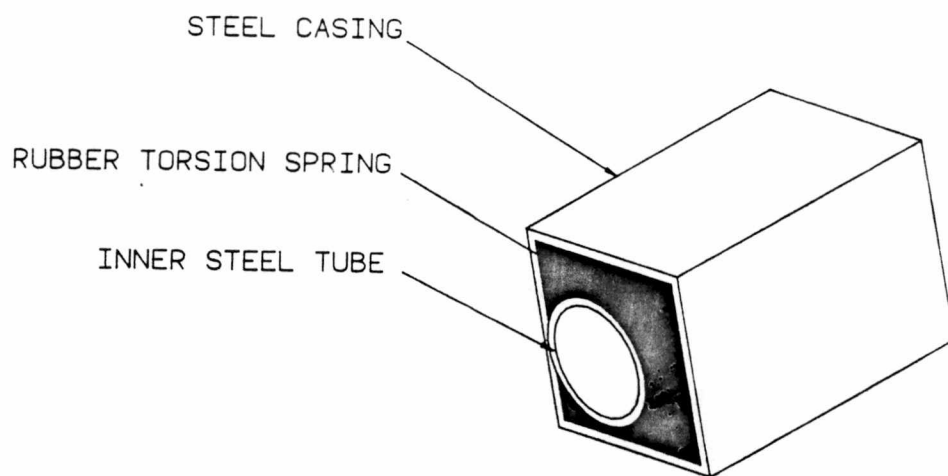


Figure 3.3. Rubber torsion spring.

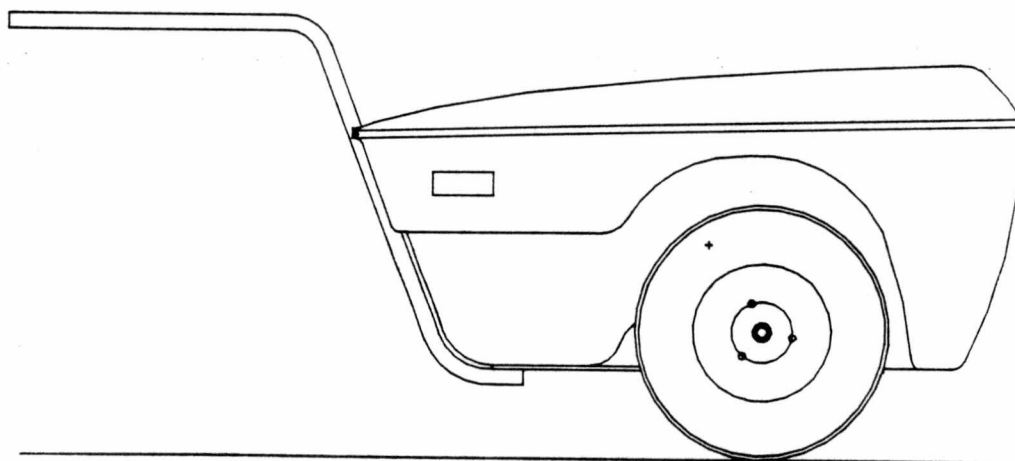


Figure 3.4. Elevated tongue configuration.

gain in forward positioning of up to twelve inches can be obtained with the gooseneck tongue. The primary disadvantage of the elevated hitch point is that, by locating the hitch point further away from the ground, low speed maneuverability may be impaired for high hitch loads. Consider the rear view of the motorcycle shown in Figure 3.5 with point C being the hitch point. When the rider leans into a turn point C rotates about the lowest point on the motorcycle tire. The trailer tongue must follow this motion since it is attached to point C. This lateral force causes the trailer to steer in the direction in which the motorcycle is leaning. Since in addition to leaning, the direction of motorcycle travel is also determined by the steering angle, the trailer may not track the motorcycle properly. This problem will be evident for large steering angles and could be significant at low speeds. In practice, however, the rider towing a trailer may learn to compensate for the problem using appropriate body motions. Thus, it is anticipated that the benefits that accrue from improved straight-line braking due to the forward location of the hitch outweigh the decrease in performance from large lean angles.

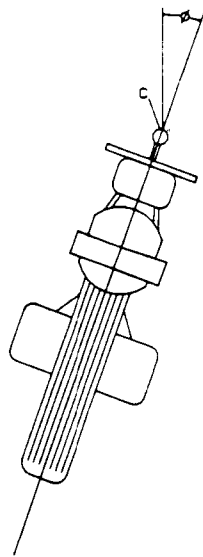


Figure 3.5. Rear view of a motorcycle with roll.

The tongue coupler to the hitch was designed to allow up to 60 degrees relative rotation from the vertical between the motorcycle and trailer. Thus, when the rider leans in turns, his actions are not resisted by torsional forces between the motorcycle and trailer. The coupler design is a quick-disconnect ball and socket device that uses the tongue itself as part of the coupling. The ball resides within an acetal copolymer plastic socket inserted into the tongue. A spring loaded outer sleeve secures the connection. The coupler has a backup retaining pin that prevents the connection from accidentally coming undone. It also provides the owner with a way of locking the trailer to the motorcycle.

3.3 Development of Conceptual Designs

With the general system configuration selected, conceptual designs were made on a computer aided design software, CADAM (Computer Augmented Design and Manufacturing). The wheels, tires, spindles, and hubs are all standard parts that can be purchased inexpensively. The hubs run on tapered roller bearings which are also included in the purchase. The conceptual design commenced with several requirements in mind:

1. Low weight components
2. Easy wheel alignment during manufacture
3. Low wear on moving parts
4. Stock dimensions to be used where possible
5. Minimum number of parts in assembly
6. Design for ease of assembly
7. Inexpensive processes

The rubber spring selected is shown in Figure 3.6. In this spring, natural rubber is bonded to a metal sleeve which contains a keyway for attachment to an inner shaft. A square steel housing fits over the spring. The torque arm is attached to the housing.

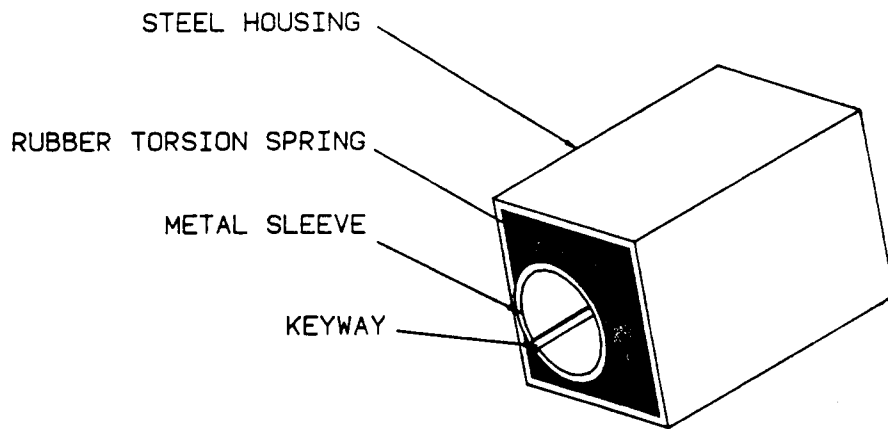


Figure 3.6. Proposed rubber torsion spring.

The first problem was to determine how the torsional loads on the rubber spring and its inner shaft would be resisted. One way of doing this would be to weld the inner shaft to a steel plate backing as shown in Figure 3.7. The steel plate would then be fastened to the trailer body. One steel plate per wheel would be required. This conceptual design was dropped for several reasons. The first reason is that the body to which the plates are attached would have to be very strong in order to resist the torsional loads arising from wheel loads. This would lead to an unnecessarily heavy trailer body. Secondly, wheel alignment would be a problem because it would depend on the alignment of body panels. It also results in two inner shafts and two steel plate assemblies.

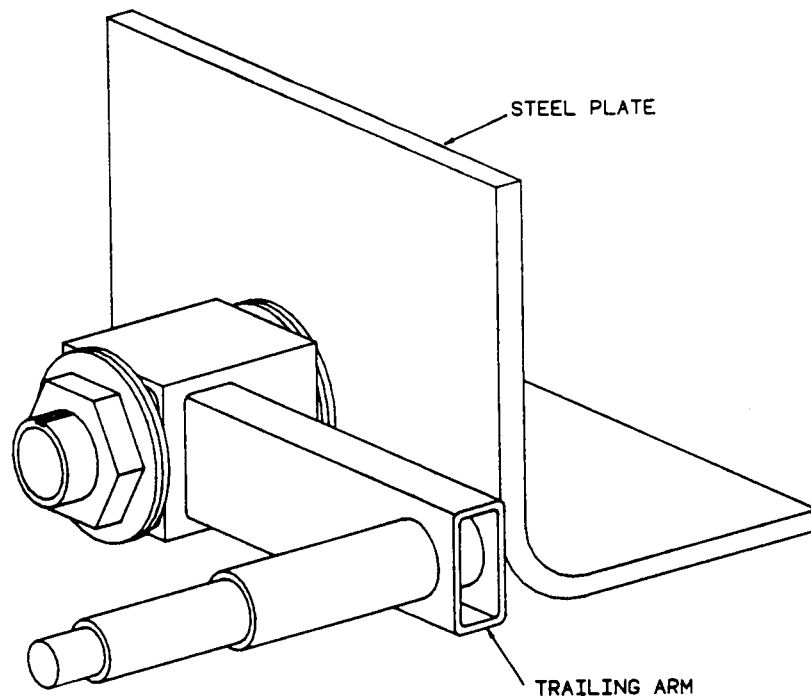


Figure 3.7. Conceptual design of trailing arm mount.

Another conceptual design involved an open ended rectangular frame to which each inner shaft would be welded, Figure 3.8. The frame would then straddle the trailer body as shown. If a flatbed is desired, this arrangement will lead to an unnecessarily high center of gravity, since the trailing arm will need a space of 4 inches above it for deflection. Also alignment would be a separate and difficult operation because after welding the frame members and inner shafts, distortions would arise due to heat affects. Also, torsional loads about the longitudinal axis of the frame would necessitate an unduly heavy frame.

The conceptual design converged on a one piece cross tube that spans the width of the trailer bed, Figure 3.9. Torsional loads would be resisted by a plate welded to the tube. The plate is bent in the middle to accomodate the axle. Bending the plate provides a suitable weld area for achieving the required strength. The tongue would be attached to the forward section of the bottom of the plate. The trailer bed would be bolted to the plate.

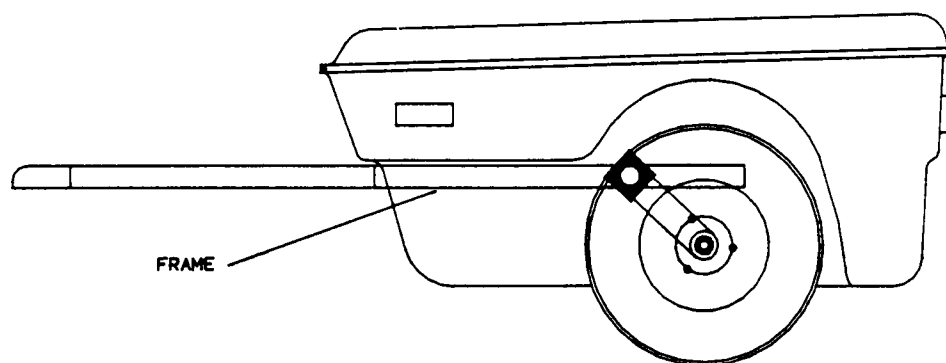


Figure 3.8. Open-ended rectangular frame design.

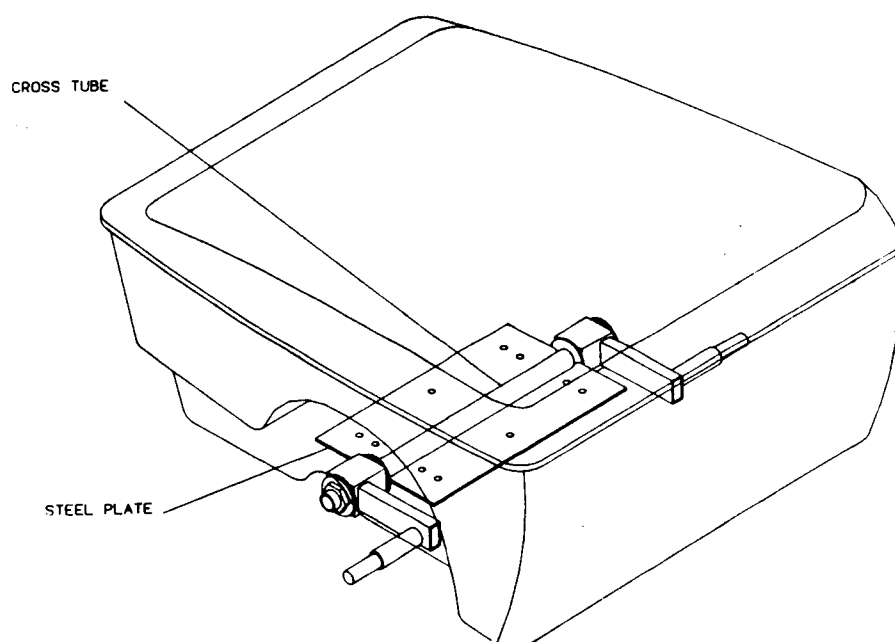


Figure 3.9. Chassis cross-tube with plate.

Several benefits arise from this arrangement. Proper wheel alignment is inherent since the resisting body is a single, straight member. The only parameter that must be monitored to achieve proper tracking is the angle between the tongue and axle as shown in Figure 3.10. The desired angle is 90 degrees. By attaching the trailer bed to the chassis plate, this design uses the trailer bed to provide additional stiffness for resisting wheel loads. Thus, a reduction in the weight of the chassis assembly can be achieved.

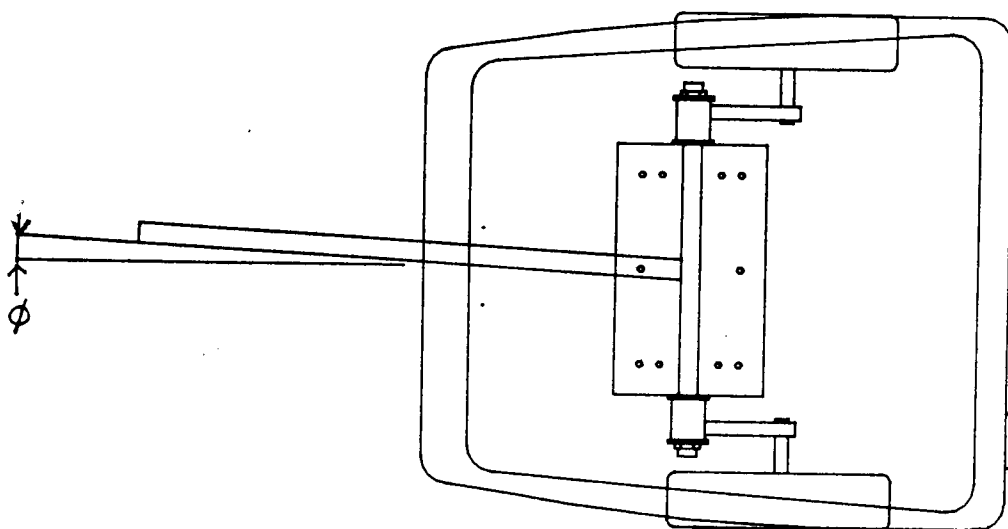


Figure 3.10. Deviation of tongue/axle angle from 90 degrees.

Another feature of this concept is that the flat plate provides a low profiled base to which any body style can be adapted without an increase in ground clearance and the corresponding elevation in the center of gravity.

The design of the coupler was based on the following requirements:

1. A 45 degree minimum relative rotation about the longitudinal axis between the trailer and the motorcycle.
2. A minimum of 25 degree pitch angle.
3. 90 degree rotation about the hitch point vertical axis.
4. The trailer must not uncouple on its own.
5. Must provide a way of securing the trailer from theft.

The coupler design stems from an original idea in which the trailer tongue is used as a socket, Figure 3.11. Within the tongue is a plastic rod that serves as the bearing surface. Since the hitch ball rides within the socket in the plastic rod, retaining pins are provided to prevent movement between the rod and the tongue. A metal sleeve slides over the tongue to capture the hitch ball when it is inside the socket. The sleeve is spring loaded to prevent the ball from disengaging. To release the tongue from the hitch ball, the spring must be compressed a certain distance that allows an opening in the sleeve to line up with the socket. A hole is provided in the tongue for a locking pin that prevents the sleeve from moving backwards and, hence, uncoupling the trailer. This also serves as a safety backup for the hitch assembly.

3.4 Failure Modes

The two most important modes of failure for the design are rubber-to-inner sleeve joint failure and the failure of the keyed connection between the shaft and the inner sleeve. The state-of-the-art in bonding rubber to metal has proven reliable for torsion springs.

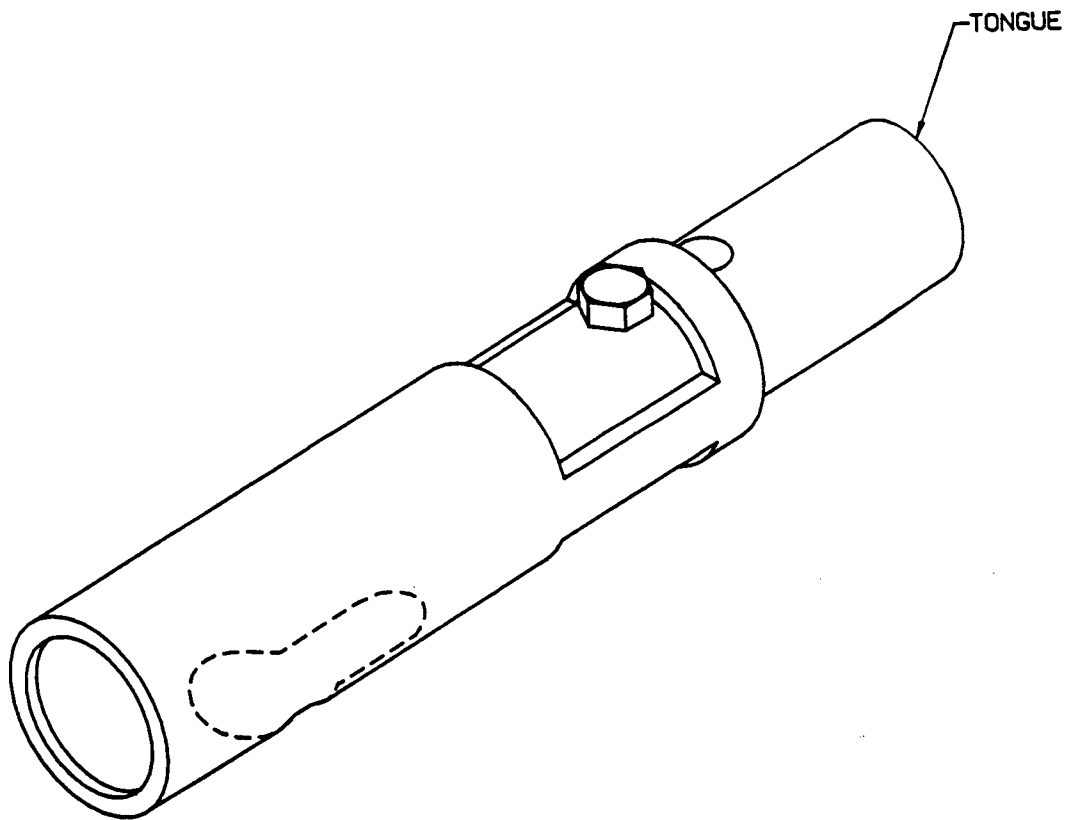


Figure 3.11. Conceptual design of coupler.

Discussions with the manufacturer's applications engineer led to the conclusion that as long as the torsional deflection is limited to 45 degrees, the joint will not fail in service. This design requires a maximum of 30 degrees rotation of the spring. Only long-term tests within the required range can verify this assertion.

The keyed connection fails due to either excessive shear stress or wear. The former can be obviated by appropriate safety factors while the latter depends on tolerances specified and achieved in manufacturing the keys. If the cross-sectional dimensions of the key produce a key that is much smaller than the keyway, then a rocking action of the key results. Over a large number of cycles, this rocking motion will lead to wear and, eventually, joint failure. Thus the keyway dimensions must be controlled in manufacturing.

Either of these two failure modes will lead to the collapse of the suspension on that side of the trailer. In the event of failure, the torsion arm will rest on its stop as shown in Figure 3.12. Since the suspension system is independent, the other wheel will not be affected and the trailer can still be towed.

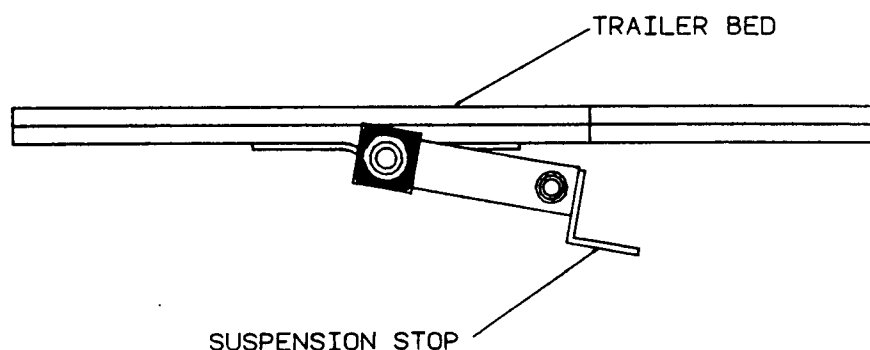


Figure 3.12. Suspension travel limiting arrangement.

3.5 Component Tests

Two components were tested before all the components necessary for an initial system test were designed. These were the rubber spring and the tire. Static and cyclic tests were performed on the rubber spring to determine if at all it could be used. The first test on

the spring was used to determine if its stiffness would drop considerably after a large number of continuous cycles. It was subjected to 10,000 cycles using a test fixture .

The rubber spring was inserted in its housing as it would be in service. An air-over-oil cylinder was used to impart a force on the trailing arm. Since the available air pressure was limited to 80 psi and the cylinder had a 1.125 inch bore, the trailing arm was made longer than it would be on the trailer. The arm would later be cut off to the desired length. The point of application of the piston force was offset from the arm using another member. This offset simulates the actual loading of the suspension. The loading produces torsional stresses on the spring as well as radial and axial stresses, Figure 3.13. Cylinder action was controlled by a programmable logic controller and produced loading in only one direction. After 10,000 cycles over a 10 hour period, the maximum rotation increased by less than 2 degrees.

Using a similar arrangement, but with a static load equivalent to the maximum suspension load, the rubber was tested for static creep. The loading was applied for 17 days. Upon removing the load, the spring showed a 3 degree set which disappeared after a load was applied for 5 cycles. These tests are by no means an adequate representation of the spring's actual life, but, along with the manufacturer's statement of possible applications, it was concluded that they could be used for the trailer springs.

3.5 Cost Reduction in Design

The most important consideration following safety in this design is the requirement of a low-cost product. The design of each component involved cost comparisons to other possible approaches, if any. The comparisons included considerations of material costs and the type and number of manufacturing steps needed to produce the component.

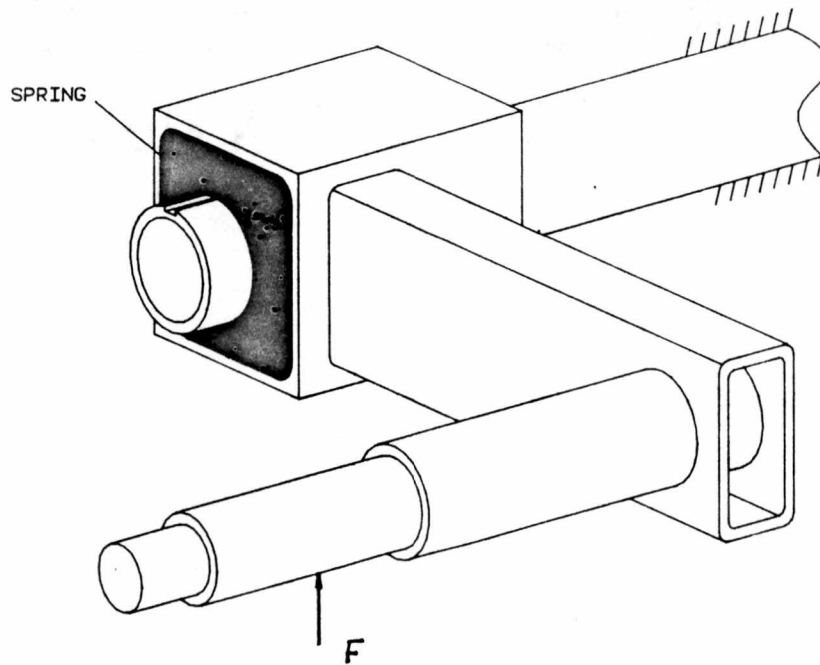


Figure 3.13. Force producing radial and axial loads on arm.

Having selected the torsional rubber spring, the design proceeded from the spring to other components. First, the trailing arm. The required housing for the rubber spring is a square steel tubing that measures 2.25 inches inside width and 3 inches length. The spring manufacturer recommends a thick wall tubing for which 0.125 inches was deemed sufficient. This tubing size is standard and can be purchased easily, cut to 3 inch lengths and the edges deburred.

The torque arm is attached to the housing. Rectangular tubing was chosen for this component for several reasons. The first reason has to do with the strength to weight ratio. The cost function in selecting the proper cross-section is weight. Consider the loading on the arm shown in Figure 3.14. For a given force F , the arm is subjected to torsion about the x-axis and bending about the z-axis. Also, lateral forces on the wheel due to cornering

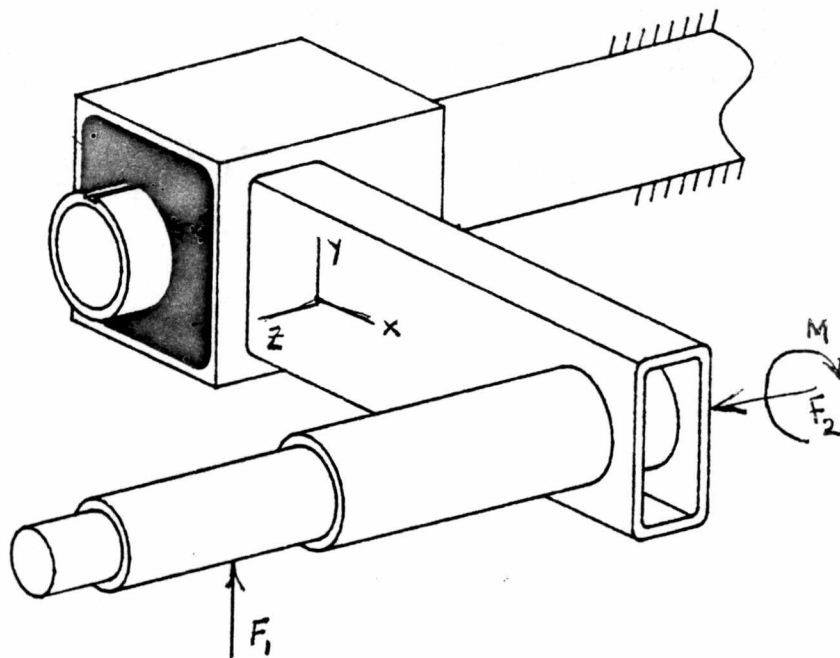


Figure 3.14. Loading of trailing arm.

produce a bending moment about the x-axis as well as additional torsion about the x-axis. The rectangular tubing provides a greater strength to weight ratio than a solid rectangular cross-section. Round tubing does not provide as good a strength to weight ratio. Because the moments of inertia about the y and z axes are the same while the force that causes a bending moment about the z axis is more than twice that producing bending about the y axis, the tubing will be unnecessarily heavy.

The attachment of the trailing arm to the cross tube posed a problem. The initial design is shown in Figure 3.15. It is necessary to that the trailing arm be prevented from sliding off the crossmember. There are two ways in which the trailing arm can come loose. One is due to the spring inner sleeve sliding out and the other is due to the spring housing coming apart from the rubber. To secure the arm, a plate would be fastened to each side of

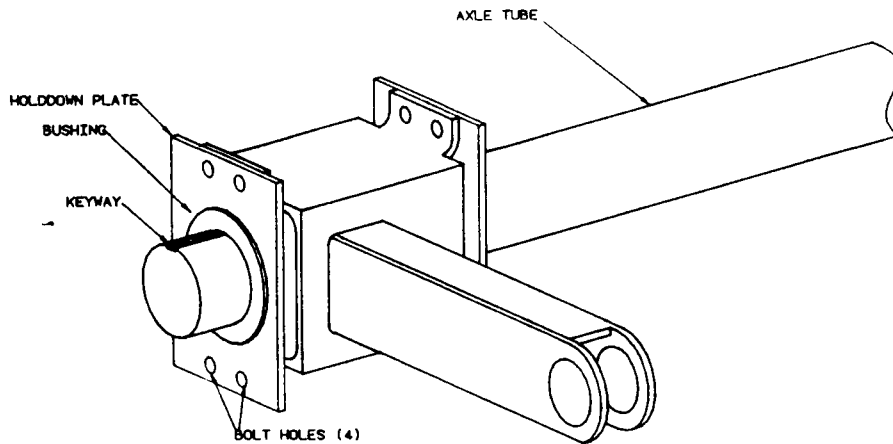


Figure 3.15. Initial spring housing design.

the spring housing. A nut that butts against the plate would be fastened to the cross tube and secured by a cotter pin. Washers will be used at the interface of the nut and the plate. The plate has an internal diameter greater than the cross tube outer diameter. Thus, when there is an axial deflection of the spring the plate would not contact the crossmember. This is important for several reasons. The first reason is that if the plate contacts the crossmember, shock loads will be transmitted directly to the cross tube necessitating greater cross-sectional properties. Secondly, because the plate can rotate relative to the crossmember, there will be considerable wear on the crossmember leading to the possibility of failure. A third reason why the plate should not come in contact with the cross tube is that a solid stop will prevent the provision of negative camber while cornering. This is considered beneficial for the lateral stability of the trailer.

This design would require two plates per arm. The plates would have at least three holes bored in them. Two of the holes would have to be threaded to accept screws. These extra manufacturing operations led to a much simpler design shown in Figure 3.16.

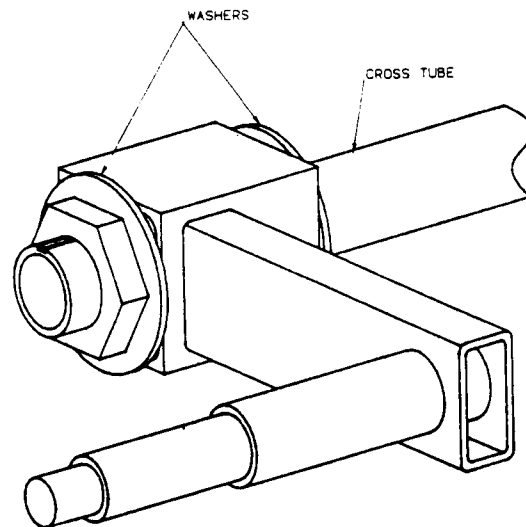


Figure 3.16. Final design of spring housing.

In this design, the plate that is secured to the trailing arm is omitted altogether. Rather, washers with a diameter as large as the 3.25 inch square spring housing are used to prevent the arm from accidentally sliding off the cross tube. One washer is installed on each side of the spring housing. This arrangement is possible because the length of the spring housing as specified by the manufacturer is shorter than the length of the spring inner sleeve. Thus the inner sleeve protrudes on both sides of the spring housing. The washers have an inner diameter that is only 0.050 inches greater than the diameter of the cross tube. Thus they can

be slipped on the crossmember. They serve the dual purpose of preventing the spring inner sleeve from sliding as well as the spring housing from disengaging from the rubber spring.

Another cost saving aspect of this design is the use of a chassis plate, Figure 3.17. It provides several functions. First, it produces torsional resistance to the wheel loads. Secondly, it serves as an edge for attaching the trailing arm to the cross tube. The arm assembly's inward movement is limited by this plate. Another function of the plate is that it serves as a base to which the trailer bed can be attached. Because it spans the entire length between trailing arms it can be relatively thin and hence of low weight.

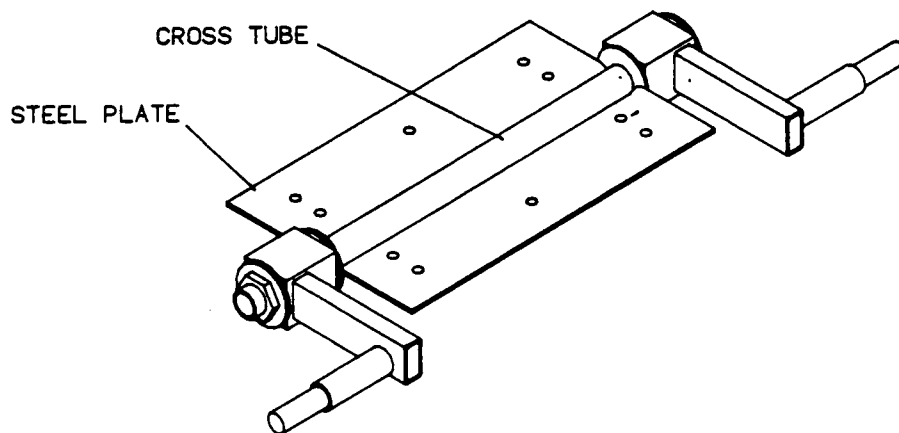


Figure 3.17. Chassis plate with cross-tube.

In coupling the trailer to the motorcycle, a coupler and a hitch are needed. The coupler involves a sleeve that slides and rotates on the trailer tongue. The socket for the hitch ball is a plastic which is inserted in the tongue. The hitch to which the ball is attached

is a two piece design. Figure 3.18 shows the arrangement. A two inch piece of rectangular tubing is welded to a base plate which is in turn bolted to the motorcycle luggage carrier rack. Because of the number of different rack designs, it is not practical to design a universal hitch.

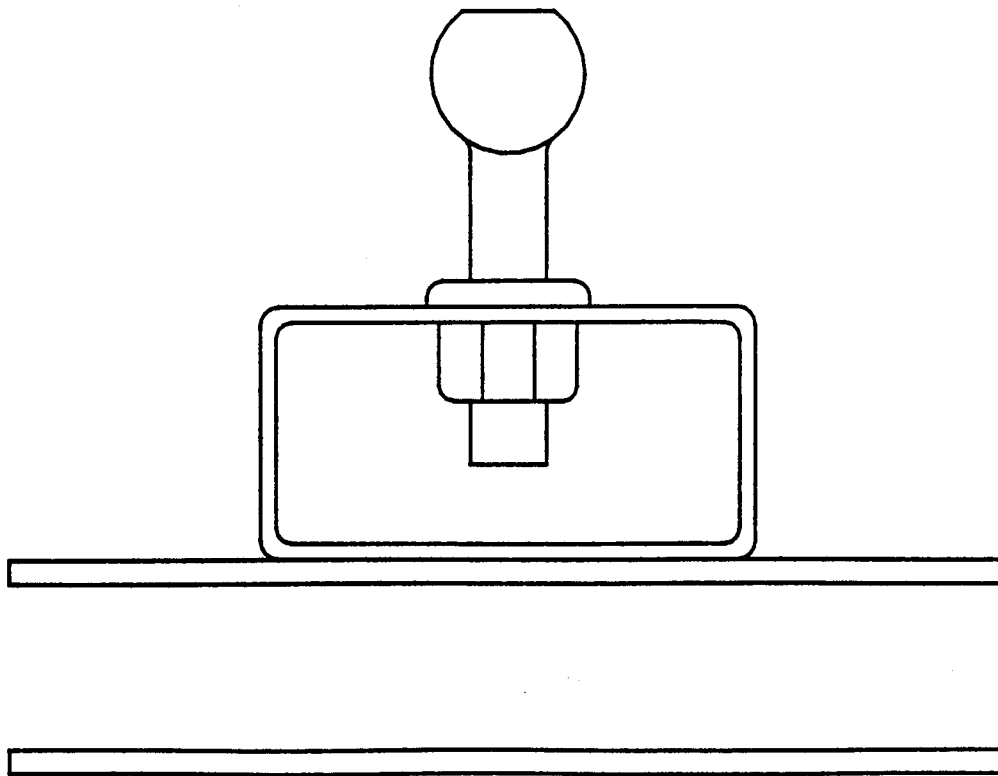


Figure 3.18. Design of trailer hitch mount.

One such design was, however, implemented with the expectation that actual production would involve several different designs for the most popular type of racks.

Welds are used extensively in this design to reduce manufacturing costs. Welding offers reduced weight over fasteners, reliable joints and reduced part complexity. It reduces costs because it substitutes for operations such as milling and drilling. It is a

relatively fast operation which does not require (in these operations) highly skilled labor.

The following joints were welded:

1. Trailing arm link to spring housing.
2. Spindle to trailing arm.
3. Arm travel stop.
4. Plate to crossmember.
5. Tongue to plate.
6. Hitch ball carrier to base plate.

All the joints listed above are critical to the operation of the trailer and the failure of any one may compromise the safety of the motorcycle/trailer combination.

CHAPTER 4

COMPONENT DESIGN AND MANUFACTURE

4.1 Trailer Wheel Loads

The suspension of a vehicle travelling over a road is subjected to a wide range of loading conditions. The vertical wheel loads will vary from values corresponding to the vehicle's static weight to up to 300 percent of its static weight. The latter occurs when the vehicle is travelling over very rough roads. Thus for a trailer gross weight of 300 lbs, the combined static plus dynamic loads is estimated to be 900 lbs assuming the worst case of zero hitch load.

The loads on the suspension produce either fatigue stresses or impact stresses. The fatigue loads arise from a vehicle travelling over roads of average roughness. Hence, alternating loads are applied to the suspension components. In fatigue analyses, the maximum vertical wheel load is given by

$$F_{z,max} = q_1 F_{z,static} \quad (4.1)$$

where q_1 is the dynamic load factor. This factor has been determined empirically [2] and is given by

$$q_1 = \frac{K_s}{F_{z,static}} + 1 \quad (4.2)$$

where K_s is the tire spring rate in N/cm and the force $F_{z,static}$ is in Newtons. For most tires, $K_s/F_{z,static}$ ranges from 0.2 to 1.5 cm^{-1} with the higher values representing high tire inflation pressures.

In addition to the vertical dynamic wheel loads, there are alternating side forces produced when the vehicle is travelling straight on rough road surfaces. As shown in Figure 4.1 a bump in the road applies a lateral force to the bottom of the tire.

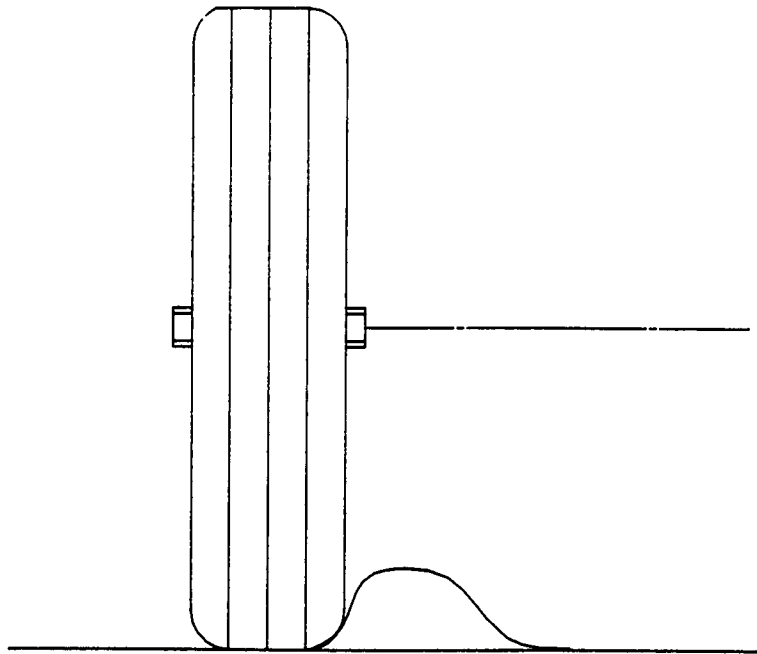


Figure 4.1. Alternating tire side force.

The alternating side force may be determined by

$$\pm F_y = q_3 F_{z,static} \quad (4.3)$$

where q_3 is the dimensionless side force factor. It is a function of road roughness and wheel loads. Empirical data show that a value of 0.3 is adequate for rough roads [2].

The impact loads occur when the vehicle's tire hits a large bump or pothole. The overload factor q_2 is usually taken to be 3.0 for very rough roads. In determining the stresses due to impact, the effect of the tires side force is also taken into account using the same overload factor.

In this design, the maximum trailer static load per wheel is 135 lbs. for stress calculations, the tires spring rate for 40 psi inflation pressure is assumed. This represents an upper limit for the tire spring rate and, consequently, an upper limit for the dynamic load factor. Tire spring rates were determined from load-deflection curves obtained from laboratory tests on a tensile tester, Figure 4.2. In the test, vertical loads were applied to the trailer tire using different tire pressures. The spring rate over the load range of interest, 50 lbs to 300 lbs, is approximately 862 lb/in (151 N/mm) at 40 psi.

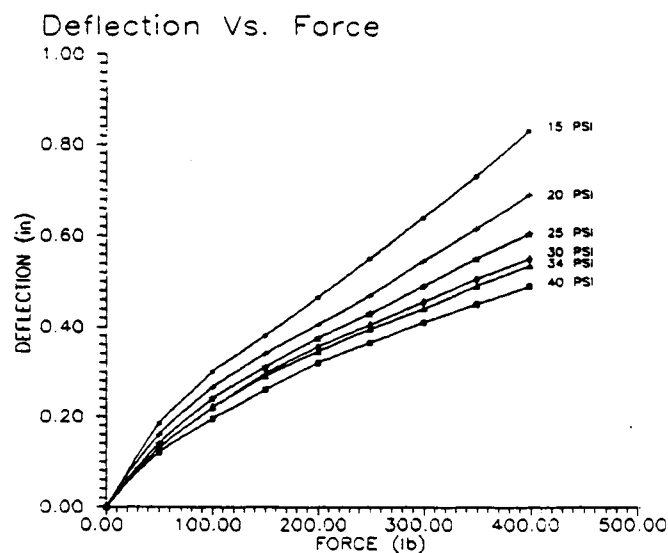


Figure 4.2. Tire load-deflection curves from laboratory test.

4.2 Component Selection

In order to minimize the cost of the trailer, the wheels, tires, hubs, spindles and bearings selected are all mass produced stock items. The wheel size selected, 480-8, is the most common trailer wheel available. With the tire mounted, the wheel diameter is approximately 16.5 inches. The wheel bearings are high speed tapered roller bearings that are sold along with standard hubs.

The rubber spring, or center bonded joint, was selected in conjunction with the swing arm dimensions. It is desirable to have the moment arm of the trailing link as long as possible. This affords a large deflection per angle of rotation. The manufacturer recommends a 45 degree limit in rotation for the spring. A 30 degree limit is chosen here as an additional factor of safety. For aesthetic reasons, the effective moment arm is limited to 7 inches. This limit keeps the suspension parts out of view.

Assuming the worst case condition in which the trailer wheels support all of the load, i.e. no hitch load, the maximum static torsional load on one rubber spring is 1050 in-lb. Based on the manufacturer's recommendation, a spring rated at 1500 in-lb at 15 degree angle of rotation was tentatively chosen. The manufacturer also recommends dynamic plus static loads no greater than 3 times the torque that produces a spring deflection of 15 degrees. This design uses an overload factor of 3 which results in a torsional load of 2625 in-lb. This is less than the 4500 in-lb stipulated. Furthermore, this spring has a radial static load rating of 1700 lbs which is about 11 times the maximum static load for this design.

The radial spring rate of the spring is important because it determines the amount of wheel camber for this design. Although some negative wheel camber is desired for lateral stability, it is important that it stay within reasonable values because of the possibility of increased tire wear. Assuming an evenly distributed radial load on the top and bottom of the spring as shown in Figure 4.3, the amount of wheel camber can be estimated. The magnitude of the distances a and b are governed by spatial considerations which limit the proximity of the suspension components to the tire.

The radial stiffness for the selected spring is stated by the manufacturer as 60,000 lb/in and was found to produce a wheel camber of 1.55 degrees with an empty trailer and 6.73 degrees for the fully loaded trailer of 300 lbs. From the analysis and a survey of camber on automobiles, the wheel camber was judged acceptable and of benefit to stability.

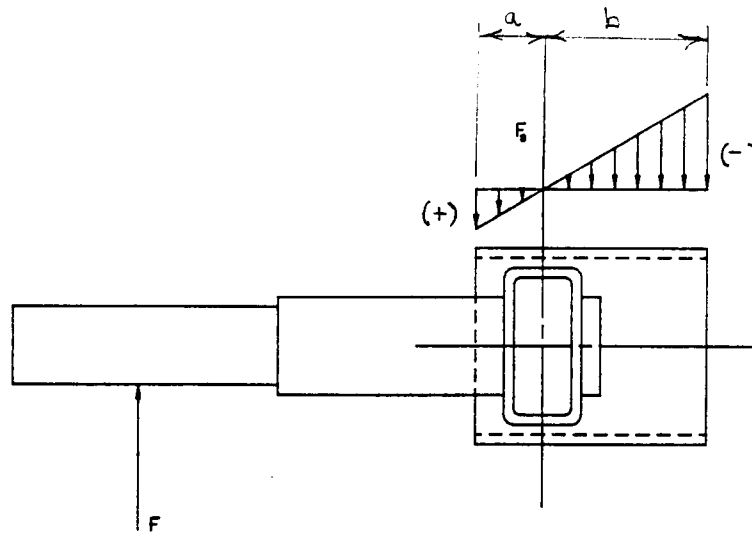


Figure 4.3. Rubber torsion spring model.

The natural frequency of the system using this rubber spring was estimated after there was some idea of the weight of components. The undamped natural frequency of the system was found using a two-degree-of-freedom model with one mass representing the wheels and the other representing trailer weight. The natural frequency for the empty trailer was estimated to be 5.9 Hz. For the fully loaded trailer, the estimate was 2.8 Hz.

4.3 Trailing Arm Design and Manufacture

Having determined that the rubber spring will be a suitable suspension component, the design of the trailing arm proceeded using the loading shown in Figure 4.4. The loads included static and dynamic forces due to the trailer weight as well as those due to cornering forces.

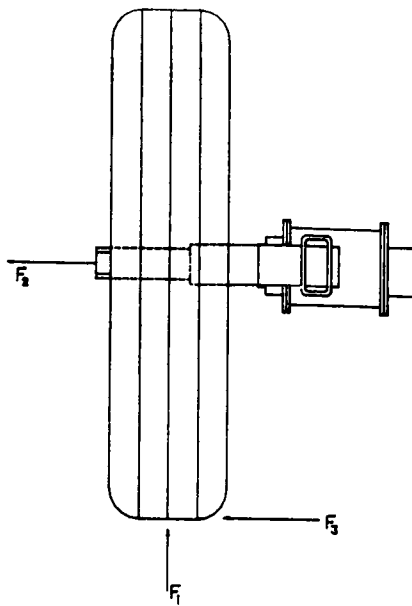


Figure 4.4. Vehicle wheel loads.

The maximum lateral acceleration for the trailer was assumed to be $0.5g$ on dry pavement. Also included in the loading is the tire side force generated when a wheel hits a bump such that a moment is produced about the longitudinal axis of the trailing link. The tire side force is taken to be 30 percent of the static vertical load. Thus, the worst case loading condition is one in which the fully loaded trailer with a high tire inflation pressure hits a bump while cornering.

There are several design constraints on the trailing link. The first constraint is that the height of the cross-section must allow the 1.25 inch spindle to be inserted into the link. Secondly, the effective length of the trailing arm must be 7 inches in accordance with the need for the longest moment arm for the given space. This limits the effective length of the link to 5.875 inches to account for the spring housing. The third constraint on the link is that the maximum cross-sectional height it can have is limited to the length available for welding on the spring housing which is 2.5 inches. Thus the link's height must be less than 2.5 inches to allow for a fillet weld all around the joint.

The constraints on the link led to the selection of a stock rectangular tube dimension of 2x1x1/8 inches. This is the minimum size available that meets the requirements listed above. The stresses in this tube were then found and compared to allowable stresses.

In the fatigue analysis, a dynamic load factor of 2.5 corresponding to high tire pressures and very rough road surfaces is used. The maximum vertical wheel load is 338 lbs with the load factor applied. The tire side force is 41 lbs while the cornering force produces an additional lateral force of 68 lbs.

In Figure 4.4, the forces F_1 , F_2 and F_3 in the directions shown combine to produce maximum torsional loads on the trailing arm. These forces produce torsional shear stresses as well as bending stresses at the joint between the link and the spring housing. The cornering force F_3 is not used in the fatigue analysis because cornering produces, for practical purposes, a steady-state loading condition. The force is used, however, in the overload analysis. Minimum loads on the link are assumed to arise from loading combinations that reduce total shear and bending stresses. Consider the directions of the forces shown in Figure 4.4. If the tire side force is reversed, it will decrease the magnitude of the couple that produces shear stresses on the link. The analysis also accounts for stress concentration due to the welded joint.

For a rectangular tube subjected to the forces being considered here, the maximum shear stress occurs at the midlength of the longer side of the cross-section and is an average value. Since the lateral forces and vertical wheel loads act along the principal centroidal axes, the maximum bending moment will lie at one of the four corners of the rectangle. The fatigue analysis is based on the maximum shear theory of failure for fluctuating loads. Appendix B shows the detailed analysis which indicates a factor of safety of 4.5 for fatigue loading of the plain carbon steel AISI 1020 CD rectangular tube.

The maximum shear theory of failure was used for overload analysis [2]. Here, the effects of stress raisers are omitted because of the high ductility of the plain carbon steels. Also, because of the ductile nature of this steel, the allowable stresses used in overload analyses are usually taken to be

$$\sigma_{all, overload} = \frac{1.2 S_{yp}}{N} \quad (4.4)$$

and

$$\tau_{all, overload} = \frac{0.58 S_{yp}}{N} \quad (4.5)$$

where N is the factor of safety.

For overload, a 300 lb weight is used to account for the possibility of a person standing or riding in the trailer. The combination of all loads that produces maximum stresses in the trailing arm link was used. The overload analysis indicated a factor of safety of 2.95 for the loading condition assumed. With this analysis it was concluded that the 2x1x1/8 inch rectangular tube meets all the requirements for the trailing arm.

Because of the possibility of spring failure and large wheel deflections it is necessary to limit the rotation of the trailing arm. A stop is provided by a piece of 3x3 inch angle iron welded to the rear of the arm. At maximum deflection the angle iron contacts the trailer bed and prevents further wheel travel. The angle iron is used instead of a bent piece of steel because the latter would increase manufacturing costs. The angle iron is simply cut into 1 inch widths - the width of the trailing arm to which it is attached. Since the tires and spring absorb most of the energy from impact loads, each trailing arm stop is assumed to be subjected to one-half of the dynamic load, or 170 lb.

4.4 Chassis Tube Design

The rubber spring to which the trailing arm is attached is held by a keyed connection between the spring inner sleeve and a hollow tube as shown in Figure 4.5. The tube is welded to a flat plate which is in turn attached to the bottom surface of the trailer bed. This arrangement resists the torsional loads produced by vertical tire forces. The rubber joint comes with a $3/16$ wide by $1/16$ inch deep keyway already cut into the inner sleeve. The keyway spans the entire length of the 3.25 inch long sleeve. Since the tube slides into the spring's inner sleeve, the tube's outer diameter is constrained to 1.375 inches. Furthermore, the tube wall thickness must be large enough to accommodate a keyway. It is desirable to use a square key because of its lower cost and ease of availability. A $3/16$ inch square key requires a 0.125 inch deep keyway in the tube. This produces another constraint

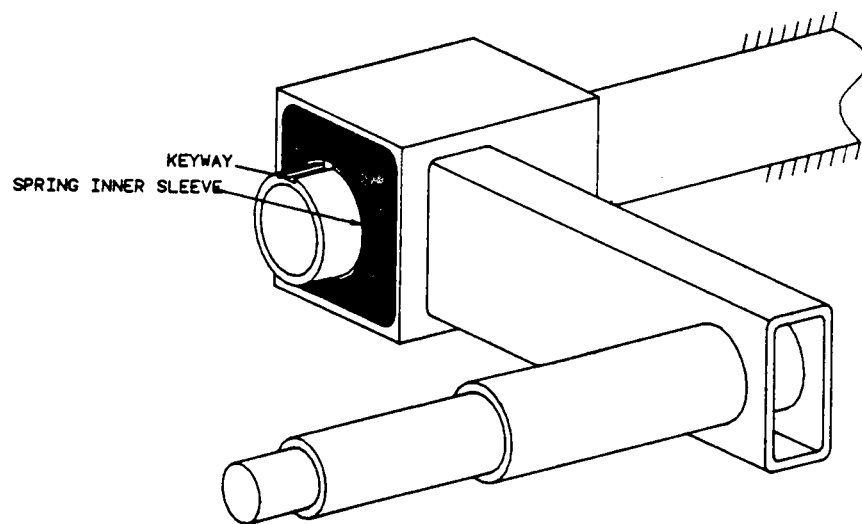


Figure 4.5. Mounting of trailing arm and spring on cross-tube.

on the tube dimensions. The tube wall thickness must be greater than 0.125 inches. Thus, the analysis involves finding the minimum wall thickness for the loading conditions.

The tube stress analysis is based on a loading similar to the one used for the trailing arm analysis. However, for the chassis tube, the dynamic load factor is much less than that of the trailing arm. This is due to the rubber spring causing a reduction in the magnitude of impact loads on the tube. Appendix A shows that the linear effective spring rate of the tire and rubber spring is 102 lb/in compared with a spring rate of 781 lb/in for the tire alone. Therefore, the dynamic load factor q_1 is found from equation 4.1 to be 1.3 whereas a factor of 2.5 was used for the trailing arm. The same overload factor of 3 is assumed for overload analysis.

The chassis tube is subjected to bending stresses and shear stresses. The tube design is based on the maximum shear theory of failure of shafts subjected to fluctuating and shock loads as well as overload. In the fatigue analysis, the maximum torsional load on the tube arises from the vertical wheel load with the dynamic load factor applied. The minimum value of torsion is assumed to be one-third of the maximum value. It is assumed that under fluctuating loads, the minimum wheel load is neither zero nor reversing. A zero wheel load indicates that the trailer is always airborne during each cycle - an unacceptable characteristic. The maximum bending loads on the tube are produced by a combination of the offset wheel and the tire side force. The minimum bending stress occurs due to a tire side force that reduces the magnitude of the total bending stress. The complete analysis of the chassis tube is presented in Appendix C.

The location of the keyway on the chassis tube is dictated by the stress distribution in the tube. Because the keyway results in a stress concentration, it is desirable to locate it in the region of minimum stress. Hence it is located on the top half of the tube cross section

as shown in Figure 4.5. This represents the region of minimum compressive and torsional stresses. The bottom half is less suitable because of the presence of tensile stresses.

4.5 Tongue Design

The configuration of trailer tongue is shown in Figure 4.6. One end is welded to the chassis of the trailer while the other is attached to an elevated hitch point. The tongue design involves an integral coupling device in which the tongue serves as a coupler member. In this design hitch ball resides within the tongue. This produces a constraint of a minimum tongue inner diameter of 1.375 inches.

For stress analysis, the loading of the trailer involves a 200 lb load concentrated at a point ahead of the wheels. This simulates a person standing in the trailer with a 200 lb weight bias towards the front of the trailer. Since, the farther forward of the trailer wheels the load concentrated, the more the tongue is stressed, this loading condition can be considered severe. The hitch load is assumed to be 10% of the gross trailer weight.

The lower curved portion of the tongue is the area of highest stress. A curved beam analysis was done on this portion with a tongue inner diameter of 1.375 inches and an assumed wall thickness of 1/16 inch. An assumption of the wall thickness is expedient because the curved beam analysis does not lend itself to the extraction of beam dimensions.

As expected the region of greater curvature experiences the higher stress. A factor of safety of 2.5 was shown by the analysis, the details of which are presented in Appendix D. However, a thicker wall of 1/8 inch was specified due to manufacturing imperfections. These imperfections arise from the bending process during which a hollow member such as the tongue assumes a non-circular cross-section after bending. The resulting oval section is not as strong as the undeformed circular cross-section. Also, some wrinkling of the member occurs in the bent region. This may lessen the strength of the curved member.

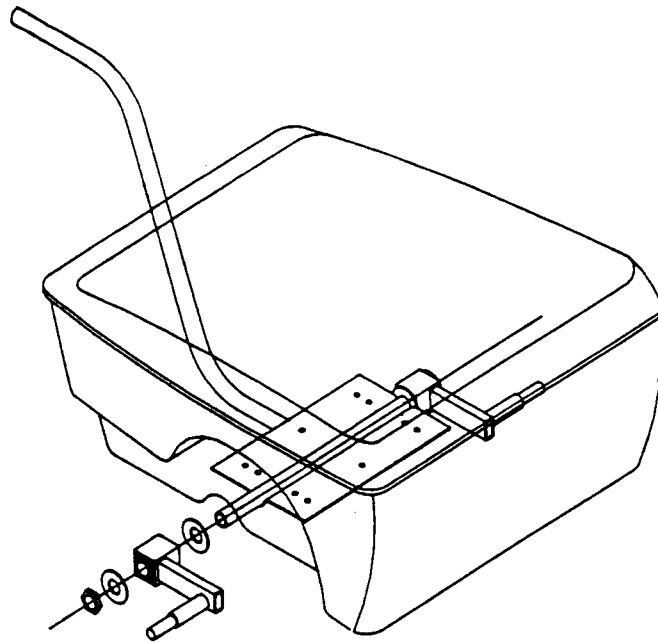


Figure 4.6. Trailer tongue arrangement.

4.6 Coupler and Hitch Design

The coupler attaches the trailer to the motorcycle by mating with a hitch ball mounted on the motorcycle. The kinematic requirements for a motorcycle/trailer coupler involves rotations about the vertical axis and the horizontal axis as well as rotation about the motorcycle's longitudinal axis. In automotive applications the maximum rotation allowed by the coupler is about 12 degrees. Because a motorcycle must lean in turns, a larger angle of rotation is required to ensure that the motorcyclist's efforts are not prematurely resisted by the coupler. Furthermore, when parked with a trailer attached, a motorcycle can accidentally tip over. If held back by the trailer coupler, damage may result to the hitch, the coupler or both.

The coupler designed here allows up to 50 degrees of relative rotation between the motorcycle and the trailer. The design is shown in Figure 4.7. It involves a sleeve that is

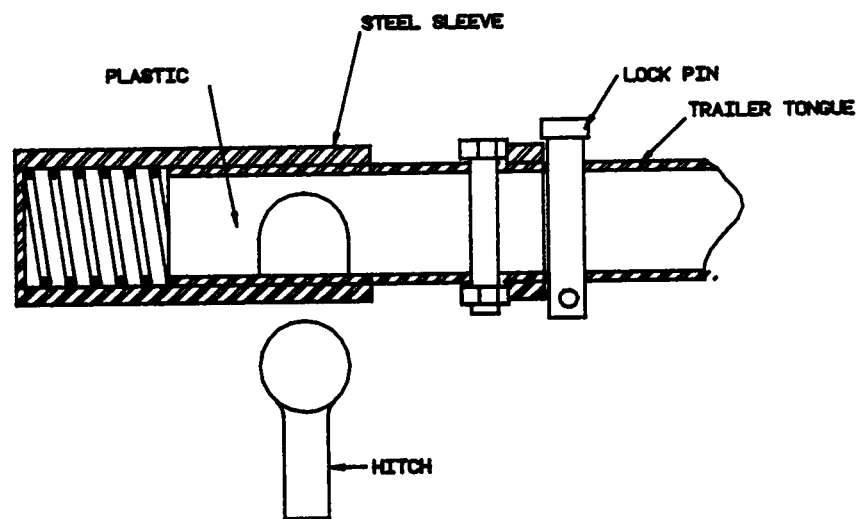
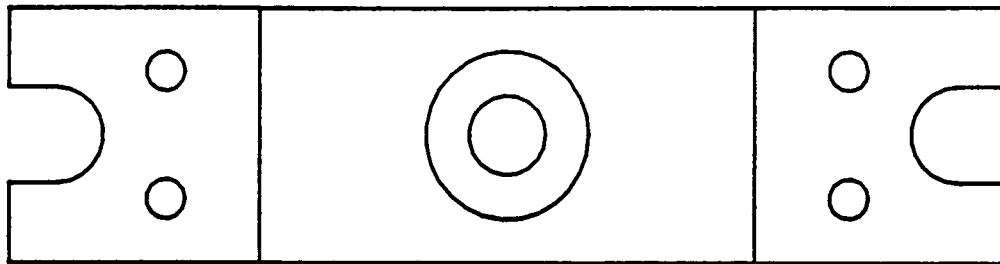


Figure 4.7. Coupler design.

free to slide and rotate on the tongue. The sleeve has an opening that allows the hitch ball to mate with the tongue thus forming a ball and socket joint. Within the tongue is a plastic socket within which the ball rotates. The sleeve has a slot that allows it to slide over the hitch ball neck. A spring at the end of the sleeve ensures that the hitch ball cannot be separated from the tongue without effort. Thus, to couple the trailer to the hitch ball, the sleeve must be pushed back to expose the opening to the socket in the tongue.

For additional safety, and as a means of securing the trailer from theft, a removable pin is inserted into a hole in the tongue at the back of the sleeve. This prevents the sleeve from sliding rearward thus preventing the connection from disengaging.

The trailer hitch design is shown in Figure 4.8. It consists of only three parts - the hitch ball, rectangular tubing and a steel bar. The hitch ball is bolted to the rectangular tubing which is in turn welded to the steel bar. The assembly is mounted on the rear luggage rack of the motorcycle. Because of the wide variety of designs of luggage racks, it is not practical to design a universal hitch. With minor variations, however, the hitch designed here can be made to fit several different racks. The manufacture of the hitch involves cutting the rectangular tubing and the steel bar to length, drilling of holes in both, and welding the two parts together.



TOP VIEW

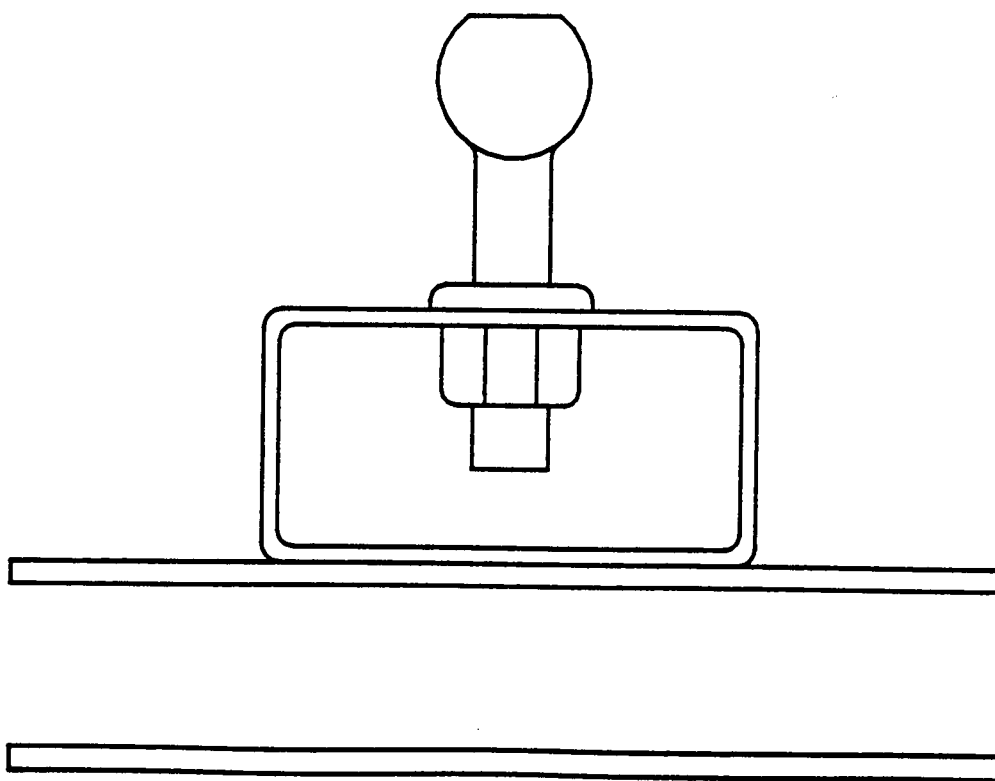


Figure 4.8. Trailer hitch design.

CHAPTER 5

EVALUATION OF PROTOTYPE

5.1 Laboratory Tests

The laboratory tests were used to determine the damping ratio and the damped natural frequency of the trailer suspension. Using an accelerometer attached to the trailer bed, the system's response to a 180 lb load was recorded on a signal analyzer. The load was suddenly removed from the trailer bed and the magnitude of the acceleration was recorded. Figure 5.1 shows the plot of the resulting motion. The ordinate of the plot is presented in microunits of gravity. The lowest value of acceleration is nonzero because of the motion transmitted by the laboratory floor. The damping ratio of the system is found by determining the logarithmic decrement of successive peaks on the plot. The logarithmic decrement δ is given by

$$\delta = \frac{1}{n} \ln \frac{x_0}{x_n}$$

For this system the damping ratio ζ can be found from its relationship to the logarithmic decrement:

$$\xi = \sqrt{\frac{\delta^2}{(4\pi^2 + \delta^2)}}$$

The experiment indicated a damping ration that varied nonlinearly. The following values were obtained using Figure 5.1:

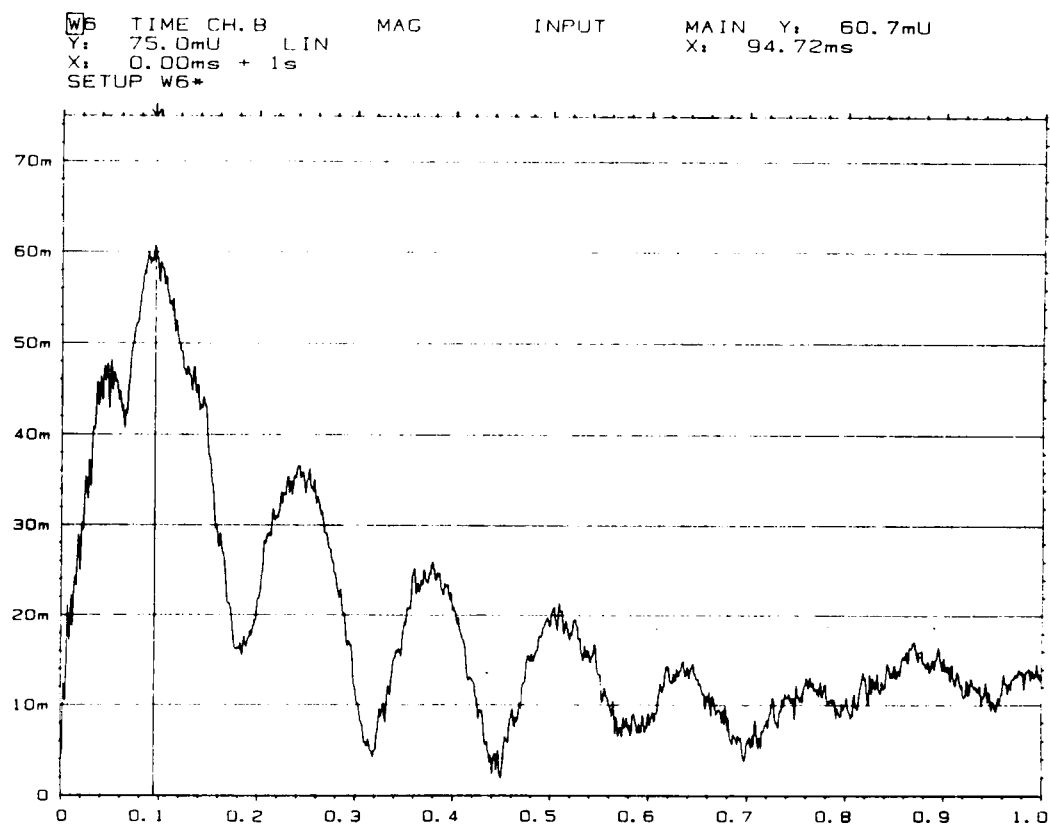


Figure 5.1. Trailer suspension system acceleration response.

n	ζ
1	0.21
2	0.20
3	0.17

It can be seen from these values that the damping ratio decreases as the number of cycles, n , increases. The natural frequency of the system can be obtained by measuring the damped period of oscillation τ from the plot in Figure 5.1. The natural frequency is given by

$$\omega_n = \frac{1}{\tau \sqrt{1 - \xi^2}}$$

From this equation, the natural frequency for the unloaded trailer - the worst case - was found to be 7.7 Hz. This value is 23.4 % larger than the 5.9 Hz estimate of the natural frequency. The discrepancy due to the fact that, unlike the analysis, the experimental test includes the effects of the tires. The higher actual frequency is worse than the estimate because it is closer to the undesirable wheel hop frequency of 12 Hz.

5.2 Road Tests

Road tests were performed to ascertain the trailer's behavior over various road surfaces as well as its influence on the maneuverability of the motorcycle. A 90 cc street-legal motorcycle was used as the tow vehicle. With a 180 lb rider, the motorcycle's combined weight was about 350 lb. Although the trailer was designed for larger motorcycles, 185 cc and above, the test motorcycle represents an extreme but probable case. A video camera was used to record most of the tests for later visual analysis.

The first performance test was used to determine an acceptable hitch load for low speed maneuverability. The low speed test involved turning and weaving at speeds less than 10 mph. It was observed that the 9 lb hitch load for the empty trailer did not have any appreciable effect on the motorcycle's handling. A 190 lb load was then placed on the trailer and the same test performed. With the load positioned such that a 45 lb hitch load (representing 15% of the gross trailer weight) was produced, a considerable degradation in maneuverability was observed. A large steering input produced oscillations in the weave mode of the motorcycle. Thus it was difficult to return the motorcycle to the straight-ahead position after negotiating a tight turn. This effect was judged unacceptable and the hitch load was reduced to 35 lb. There was a definite improvement in the handling of the combination vehicle. Further reductions in the hitch load led to the conclusion that a 25 lb load represents the acceptable limit for the test motorcycle. This is about 8% of the gross trailer weight.

The next test performed on the trailer involved its response to a washboard surface simulated by placing 2x4 inch pieces of wood one foot apart over a 7 ft distance. The motorcycle and trailer were ridden over this setup at speeds varying from 5 mph to 30 mph. The test was run both with the two trailer wheels running over the rough surface as well as with one trailer wheel running on the smooth road surface. Also a straight-ahead direction of travel as well as turning maneuver were used to observe the trailer's behavior over the washboard surface.

In this test, tire pressures played an active role. It was observed that a 40 psi tire pressure caused the trailer to bounce off the ground a distance of about 1 foot. However, once the trailer wheels contacted the road surface, the oscillations were quickly damped out by the trailer's rubber springs. The tire stiffness was reduced by lowering the pressure in 5 psi increments and observing its effect. A 20 psi pressure was judged to produce an

acceptable amount of bounce. In all cases the forces transmitted to the motorcycle by riding the trailer over the rough surface was minimal.

Further tests on the motorcycle and trailer combination involved braking tests while travelling straight and while cornering. Also an assessment of the trailer's effect on moderate speed performance was made. The trailer did not cause the motorcycle to go unstable while braking either in the straight direction of travel or while turning. Comparisons were made between the motorcycle with the trailer attached and without. Weaving at a moderate speed of 35 mph on asphalt did not produce any unpleasant effects to the rider. The trailer consistently tracked the motorcycle and did not oversteer enough to be noticeable. With the load positioned to produce an 8% hitch load, lateral oscillations - trailer sway, did not pose a problem.

The last test performed on the trailer was that of the swivel coupler. The motorcycle was tipped over to see if the coupler would allow it to fall independent of the trailer without sustaining any damage. The coupler performed as expected. It allowed the motorcycle to lean until it touched the ground without being restrained by the trailer. The coupler, however, developed a binding problem in which the sleeve did not slide freely as designed. The binding was caused by plastic deformation of the sleeve at its weakest region - the opening that allows the hitch ball to mate with the socket.

5.3 Summary of Design

The following is a summary of the design of the motorcycle cargo trailer and the results obtained from tests performed:

1. A low weight of 98 lbs was achieved. The trailer can be towed by small motorcycles in the 125 to 250 cc range.

2. The ground clearance obtained in the design is 4 inches lower than in conventional trailers.
3. The handling, stability and braking performance of the motorcycle are not impaired by this design.
4. Use of rubber springs produces significantly fewer oscillations than in the conventional trailer.
5. The trailing arm suspension system results in a roll center located at ground level compared to 15 inches above ground level for conventional trailers.
6. A 6 degree negative camber is obtained under fully loaded conditions due to the arrangement of the suspension system. This is beneficial to lateral stability.
7. The flat trailer bed allows the use of a modular approach for different body styles.
8. The coupler design allows a total of 120 degrees of relative rotation between the motorcycle and the trailer.
9. Only standard manufacturing operations such as milling, drilling, and bending are required for building the trailer. Also, the assembly requires only basic hand tools.
10. The design involves only stock parts and dimensions.
11. The cost of building the trailer is estimated as \$130 dollars based on U.S. labor costs at the time of development. A breakdown of the cost is presented in Appendix E.

CHAPTER 6

RECOMMENDATIONS

There are several areas of the design of the trailer that can be improved. These improvements will result in lower manufacturing costs and increased safety. The first area of improvement is in the rubber torsion spring arrangement. Towards the end of this development, another manufacturer of rubber springs, Lovejoy Incorporated, was located. The company offers rubber torsion springs that have a square inner sleeve instead of the round one used in this design. Thus, a keyed connection is not needed to secure the spring's inner sleeve to the crosstube. A square crosstube could be used to provide the necessary torsional resistance. This arrangement will result in a less expensive crosstube because the keyway operation is avoided.

For safety reasons, it is better to use a straight member for the trailer tongue. This modification arises from the possibility of injury to the rider in the event of another vehicle strikes the trailer from the rear. The force of impact could break the hitch mount and allow the elevated tongue to cause injury to the rider. The straight tongue increases safety and lowers manufacturing costs due to the absence of the bending process.

Further tests must be made to determine the life of the rubber spring under actual loading conditions. Also, as is done for automobile structures, fatigue tests should be conducted on structural members to ensure adequate life.

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APPENDIXES

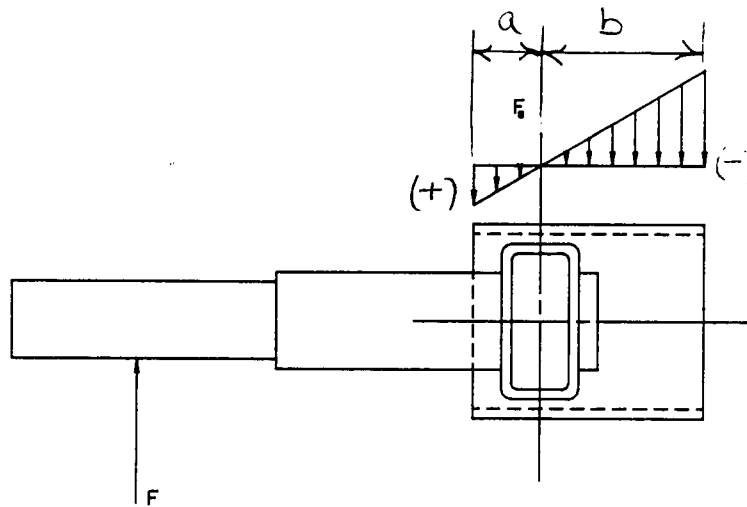
APPENDIX A

RUBBER SPRING ANALYSIS

APPENDIX A

RUBBER SPRING ANALYSIS

The amount of wheel camber due to the radial deflection of the rubber spring was found by using the model shown in the figure below. An evenly distributed load along the length of the spring was assumed.



Assuming the resisting forces to be concentrated at the centroid of the triangular load distributions and summing moments about point O,

$$M_a = F_1 a + F_2 b$$

Since $F = kx$,

$$M_a = kx_1 a + kx_2 b$$

From similar triangles,

$$x_2 = \frac{b}{a} x_1$$

Thus,

$$M_a = kx_1 \left(a + \frac{b^2}{a} \right)$$

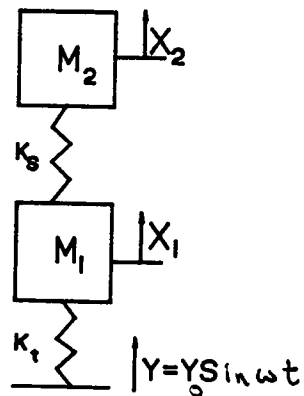
The camber θ_c is given by

$$\theta_c = \sin^{-1} \left(\frac{x_1}{a} \right)$$

Substituting for θ_c and solving produces

$$\theta_c = \sin^{-1} \left[\frac{M_a}{k(a^2 + b^2)} \right]$$

The undamped natural frequency of the trailer can be estimated using a two-degree-of-freedom model where M_1 and M_2 represent the mass of the tires and the mass trailer mass respectively. k_s and k_t are the spring stiffness and tire stiffness.



The forcing function y is assumed to have a motion represented by $y_0 \sin \omega t$. The equations of motion for this system are

$$m_1 \ddot{x}_1 + k_t x_1 - k_s x_2 + k_s x_1 = k_t y$$

$$m_2 \ddot{x}_2 + k_s x_2 - k_s x_1 = 0$$

The steady-state solution to the equation is desired. For the assumed motion,

$$x_1 = X_1 \sin \omega t$$

$$x_2 = X_2 \sin \omega t$$

Substituting these and their derivatives into eqn. produces

$$(-m_1 \omega^2 + k_t + k_s) X_1 - k_s X_2 = k_t y_0$$

$$-k_s X_1 + (-m_2 \omega^2 + k_s) X_2 = 0$$

The natural frequency is found by setting the determinant of this system of equations to zero, i.e.,

$$\begin{vmatrix} -m_1 \omega^2 + k_t + k_s & -k_s \\ -k_s & -m_2 \omega^2 + k_s \end{vmatrix} = 0$$

Thus

$$(-m_1 \omega^2 + k_t + k_s)(-m_2 \omega^2 + k_s) - k_s^2 = 0$$

Combining terms and simplifying in terms of ω leads to

$$\omega^4 - \left(\frac{k_2}{m_2} + \frac{k_t + k_s}{m_1} \right) \omega^2 + \frac{k_s k_t}{m_2 m_1} = 0$$

Let

$$\lambda = \omega^2$$

$$\alpha = \frac{k_s}{m_2} + \frac{k_s + k_t}{m_1}$$

$$\beta = \frac{k_t k_s}{m_1 m_2}$$

Thus,

$$\lambda = \frac{\alpha}{2} \pm \left(\frac{\alpha^2 - 4\beta}{2} \right)^{\frac{1}{2}}$$

The values used for the variables are as follows:

Unsprung mass, $m_1 = 42/386.4 \text{ lb s}^2/\text{in}$

Empty trailer mass, $m_2 = 57/386.4 \text{ lb s}^2/\text{in}$

Tire stiffness at 30 psi, $k_t = (2)781 = 1562 \text{ lb/in}$

Tire stiffness values are obtained from the load-deflection curves of Figure 4.2.

For the torsion spring the linear deflection δ is given by

$$\delta = L \sin \Theta$$

where Θ is the angular displacement of the spring and is given by

$$\theta = \frac{PL}{k_{s\theta}}$$

$k_{s\theta}$ is the spring torsional stiffness in in-lb/rad.

Assuming small angles,

$$\delta = \frac{PL^2}{k_{s\theta}} = \frac{P}{k_s}$$

Therefore, k_s is the linear spring stiffness in lb/in.

APPENDIX B

TRAILING ARM ANALYSIS

APPENDIX B

TRAILING ARM ANALYSIS

Stress Analysis

The loads used for the analysis of the trailing arm are shown in the figure below. The maximum shear stress theory of failure for fluctuating loads is used for the fatigue analysis.

The values used in the analysis are as follows:

Maximum Lateral Acceleration, $a_c = 0.5g$

Tire Side Force, $F_1 = 0.3(135) = 40.5 \text{ lbs}$

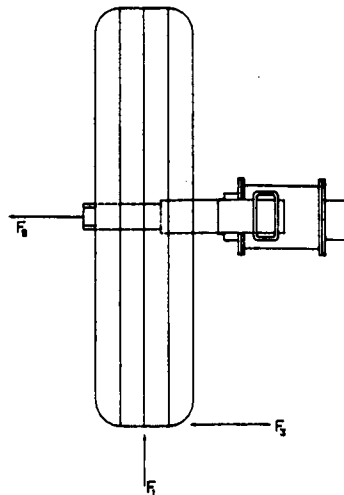
Maximum Vertical Wheel Load, $F_2 = 135 \text{ lbs}$

Tire Cornering Force, $F_3 = 0.5(135) = 67.5 \text{ lbs}$

Wheel Radius, $s_1 = 8.25 \text{ inches}$

Wheel Offset from Trailing Arm Axis, $s_2 = 5 \text{ inches}$

Effective Link Length, $s_3 = 5.875 \text{ inches}$



Wheel loads

For fatigue analysis, the loads of interest are the moments caused by the tire side force F_1 and the vertical wheel load F_2 . The maximum torsion is

$$T_{max} = F_1 s_1 + q_1 F_2 s_2$$

where q_1 is the dynamic load factor assumed to be 2.5. Thus the maximum torsion on the tube is 2022 in-lb. Minimum torsion occurs at the static vertical load on the wheel minus a tire side force that reduces the moment on the trailing arm, i.e.

$$T_{min} = F_2 s_2 - F_1 s_1$$

The minimum torsion is 341 in-lb. Similarly, the maximum bending moment is given by

$$M_{max} = q_1 F_2 s_3$$

which produces a value of 1983 in-lb. The minimum value is the static load which is 793 in-lb.

The average shear stress for the rectangular tube is given in [6] as

$$\tau_{avg} = \frac{T}{2 t (a - t) (b - t)}$$

and it occurs at the middle of the longer side of the rectangle. The bending stress is

$$\sigma_x = \frac{M a}{2 I_x}$$

The maximum shear theory of failure for combined bending and torsion [7] produces a maximum shear stress of

$$\tau_{max} = \frac{0.5 S_{yp}}{N} = \left[\left(\frac{\sigma_x}{2} \right)^2 + \tau_{avg}^2 \right]^{\frac{1}{2}}$$

where N is the factor of safety and S_{yp} is the tensile yield strength of the material. Substituting for σ_x and τ_{avg} , the Soderberg equation [7], yields

$$\tau_{max} = \frac{0.5 S_{yp}}{N} = \left[\frac{1}{4} \left(\sigma_m + \frac{S_{yp}}{S_e} \sigma_r \right)^2 + \left(\tau_{avg,m} + \frac{S_{syp}}{S_{es}} \tau_{avg,r} \right)^2 \right]^{\frac{1}{2}}$$

In this analysis, the mean stresses are taken to be the average of the maximum and minimum stresses. The only applicable stress concentration factor arises from welding. For the 2x1x1/8 inch rectangular tube, the stresses were found as follows:

$$\tau_{avg} = \frac{T}{2 t (a - t) (b - t)}$$

$$\tau_{avg,max} = \frac{2022 \text{ in-lb}}{2 (0.125) (2 - 0.125) (1 - 0.125) \text{ in}^3} = 4930 \text{ psi}$$

$$\tau_{avg,min} = \frac{341 \text{ in-lb}}{2 (0.125) (2 - 0.125) (1 - 0.125) \text{ in}^3} = 831 \text{ psi}$$

Therefore,

$$\tau_{avg,m} = \frac{4930 + 831}{2} = 2881 \text{ psi}$$

$$\tau_{avg,r} = \frac{4930 - 831}{2} = 2050 \text{ psi}$$

The moments of inertia about the horizontal and vertical axes of the link are

$$I_x = \frac{1}{12} \left[(1)(2)^3 - (1 - 2(0.125))(2 - 2(0.125))^3 \right] = 0.3317 \text{ in}^4$$

$$I_y = \frac{1}{12} \left[(2)(1)^3 - (2 - 2(0.125))(1 - 2(0.125))^3 \right] = 0.1051 \text{ in}^4$$

The bending moments are now found as follows:

$$\sigma_{x,max} = \frac{M_{max} a}{2 I_x}$$

$$\sigma_{x,max} = \frac{1983 \text{ in-lb} (1 \text{ in})}{2 (0.3317 \text{ in}^4)} = 2989 \text{ psi}$$

$$\sigma_{x,min} = \frac{M_{min} a}{2 I_x}$$

$$\sigma_{x,min} = \frac{793 \text{ in-lb} (1 \text{ in})}{2 (0.3317 \text{ in}^4)} = 1195 \text{ psi}$$

Hence

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} = \frac{2989 + 1195}{2} = 2092 \text{ psi}$$

$$\sigma_m = \frac{\sigma_{max} - \sigma_{min}}{2} = \frac{2989 - 1195}{2} = 897 \text{ psi}$$

For plain carbon steel, AISI 1020 CD [7],

$$S_{yp} = 66,000 \text{ psi}$$

$$S_{syp} = 0.58 S_{yp} = 38,280 \text{ psi}$$

$$S_n' = 0.5 S_u = 0.5 (78,000) = 39,000 \text{ psi}$$

$$S_{sn}' = 0.29 S_u = 22,620 \text{ psi}$$

$$S_e = C_w S_n' = 0.833 (39,000) = 32,487 \text{ psi}$$

$$S_{es} = C_w S_{sn}' = 0.833 (22,620) = 18,842 \text{ psi}$$

Solving the Soderberg equation for N and substituting values produces

$$N = 0.5 (66000) \left[\frac{1}{4} \left(2092 + \frac{66000}{32487} (897) \right)^2 + \left(2881 + \frac{38280}{18842} (2050) \right)^2 \right]^{-\frac{1}{2}} = 4.51$$

For overload considerations, the following forces were used:

Total Weight on Wheels = 400 lbs

Tire Side Force, $F_1 = 0.3(200) = 60$ lbs

Maximum Vertical Wheel Load, $F_2 = 200$ lbs

Tire Cornering Force, $F_3 = 0.5(200) = 100$ lbs

The overload factor for the vertical wheel load, q_3 , is taken to be 3. The stresses of interest are the maximum shear stress due to torsion and the maximum bending stresses due to vertical loads and lateral loads. The maximum torsion is obtained from the sum of moments produced by three couples. The first is a 3000 in-lb moment due to the vertical force F_2 with the overload factor applied. The second moment is due to the tire side force and is 495 in-lb. Thirdly, the tire cornering force produces a 825 in-lb moment. Thus, the maximum torsion on the link is 4320 in-lb.

Two moments - one due to the vertical force and the other due to the lateral force - are produced. These are

$$M_1 = (F_1 + F_3) s_3 = (160)(5.875) = 940 \text{ in-lb}$$

$$M_2 = q_3 F_2 s_3 = (3)(200)(5.875) = 3525 \text{ in-lb}$$

The stresses were then found from these loads.

Shear stress:

$$\tau_{avg, max} = \frac{4320 \text{ in-lb}}{2(0.125)(2 - 0.125)(1 - 0.125) \text{ in}^3} = 10533 \text{ psi}$$

Bending stress due to F_1 and F_3 :

$$\sigma_1 = \frac{M_1 c_1}{I_y} = \frac{940(0.5)}{0.1051} = 4470 \text{ psi}$$

$$\sigma_2 = \frac{M_2 c_2}{I_x} = \frac{3525(1)}{0.3317} = 10627 \text{ psi}$$

Since the forces F_1 , F_2 and F_3 act along the principal centroidal axes, the total bending stress is the sum of the individual stresses. Thus

$$\sigma_{total} = \sigma_1 + \sigma_2$$

$$\sigma_{total} = 10627 + 4470 = 15097 \text{ psi}$$

where σ_{total} occurs at one of the corners of the tube.

From the maximum shear stress theory of failure,

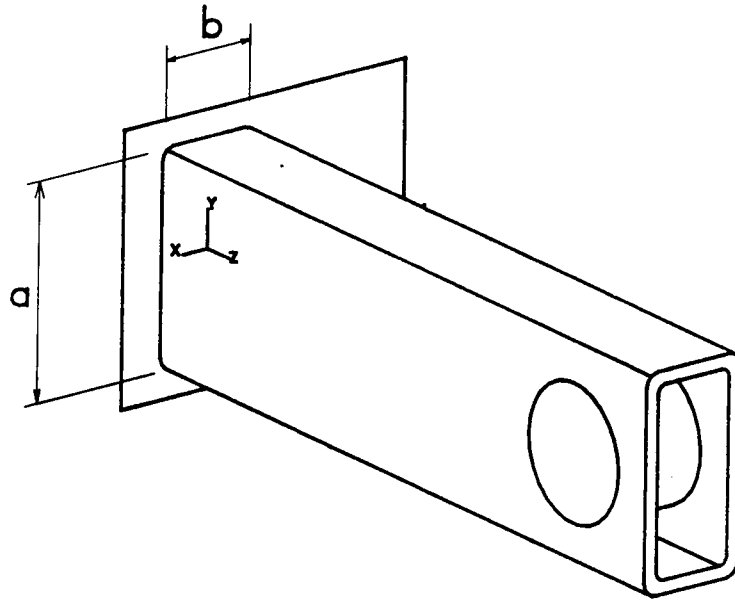
$$\tau_{max} = \left(\frac{1}{4} \sigma_{total}^2 + \tau_{avg}^2 \right)^{\frac{1}{2}}$$

$$\tau_{max} = \left(\frac{1}{4} (15097)^2 + (10533)^2 \right)^{\frac{1}{2}} = 12959 \text{ psi}$$

For AISI 1020 CD steel, the yield strength in shear is 38280 psi. Hence, the stresses due to overload indicate a factor of safety of 2.95.

Determination of Weld Size

The required type of weld for the joint is a fillet weld. The size of the weld was determined by treating the weld as a line. Thus, forces per lineal inch, and not stresses, are analyzed. For a rectangular cross-section with sides a and b in the figure on the next page, the section modulus S_w and the polar moment of inertia J_w used for welds are presented in [8] as



Trailing arm weld area.

$$S_w = ab + \frac{a^2}{3}$$

and

$$J_w = \frac{(a+b)^3}{6}$$

Where S_w is for bending about the x-axis.

The force per linear inch on the weld due to bending is

$$f = \frac{M}{S_w}$$

where M is the applied moment in in-lb. For torsion, the force is

$$f_T = \frac{Tb}{J_w}$$

where, for this non-circular cross-section, the value of the smaller side is used. Using the wheel loads for the overload condition, the force on the weld due to bending about the x-axis is

$$f_1 = \frac{q_3 M_1}{S_{w_1}} = \frac{q_3 M_1}{ab + \frac{a^2}{3}} = \frac{(3)(200)(5.875)}{(2)(1) + \frac{2^2}{3}} = 1058 \text{ lb/in}$$

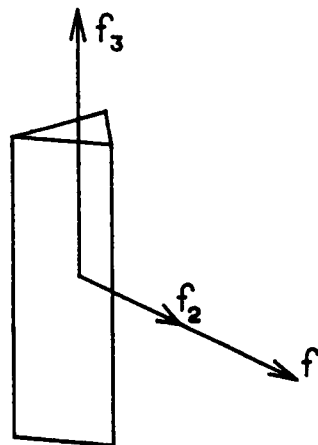
Similarly, for bending about the y-axis,

$$f_2 = \frac{M_2}{S_{w_2}} = \frac{M_1}{ab + \frac{b^2}{3}} = \frac{(160)(5.875)}{(1)(2) + \frac{1^2}{3}} = 403 \text{ lb/in}$$

A 4320 in-lb torsional load is applied to the weld thus producing

$$f_T = \frac{Tb}{J_w} = \frac{Tb}{\frac{(a+b)^3}{6}} = \frac{(6)(4320)(1)}{(1+2)^3} = 960 \text{ lb/in}$$

These forces are shown acting on the weld in the figure below.



Forces acting on weld.

As in the stress analysis of the rectangular tube, the locations of the maximum shear stress and the maximum bending stress are assumed to be the same. The forces on the weld are then added vectorially to obtain the resultant force f_r

$$f_r = \left((f_I + f_2)^2 + f_T^2 \right)^{\frac{1}{2}} = 1748 \text{ lb/in}$$

The allowable fatigue stress for the fillet weld is taken to be 27 percent of the minimum tensile strength of the weld metal. The weld metal is an E60XX and has a minimum tensile strength of 62000 psi. The weld size w is determined from the ratio of the resultant force to the allowable fatigue stress. This ratio indicates a weld size of 0.1 inch. The specified weld size is 0.25 in. This results in an additional factor of safety of 2.5.

APPENDIX C

CHASSIS TUBE STRESS ANALYSIS

APPENDIX C

CHASSIS TUBE STRESS ANALYSIS

Before finding the required diameter of the chassis tube, it is necessary to determine the size of the keyway on the tube. Since the keyway that comes with the rubber spring inner sleeve is 3/16x1/16x3.25 inches, all that is necessary is a confirmation that the keyway will be suitable for this application.

Assuming that the resistive forces on the key are located on the tube outer diameter, the transmitted torque is

$$T = \frac{FD}{2}$$

For shear stress on a plane tangent to the tube circumference, the design shear stress is

$$S_s = \frac{F}{wL}$$

Thus the transmitted torque is given by

$$T = \frac{S_s wLD}{2}$$

Based on compression the torque is

$$T = \frac{S_c wLD}{4}$$

The minimum key length can be found by assuming a key made from AISI 1020 CD steel with a yield point of 66,000 psi and a shock load factor of 4. Thus

$$S_s = \frac{0.58 S_{yp}}{N} = 9570 \text{ psi}$$

The key length based on shear stress considerations for a maximum torsion overload of 3000 in-lb is

$$L = \frac{2T}{S_s w D} = \frac{2(3000)}{(9570)(.188)(1.375)} = 2.43 \text{ in}$$

Thus a key length of 3.25 inches more than satisfies the shear stress requirements.

For bearing stresses a factor of safety of 2 is used:

$$L = \frac{4T}{S_c w D} = \frac{4(3000)}{(33000)(.188)(1.375)} = 1.4 \text{ in}$$

The preceding analysis verifies that the 3/16 inch square key with a length of 3.25 inches meets the stress requirements.

The loading of the chassis tube in operation produces an area of maximum stress that lies on the tube outer diameter. The values used for the analysis are as follows:

Maximum Lateral Acceleration, $a_c = 0.5g$

Tire Side Force, $F_1 = 0.3(135) = 40.5 \text{ lbs}$

Maximum Vertical Wheel Load, $F_2 = 135 \text{ lbs}$

Tire Cornering Force, $F_3 = 0.5(135) = 67.5 \text{ lbs}$

Wheel Radius, $s_1 = 8.25$ inches

Wheel Offset to point A, $s_2 = 7.5$ inches

Link Length, $s_3 = 7$ inches

The torsional load for fatigue analysis on the tube arises solely from vertical wheel loads with a maximum value given by

$$T_{max} = q_1 F_2 s_3 = (1.3)(135)(7) = 1229 \text{ in-lb}$$

Its minimum value is assumed to be

$$T_{min} = \frac{1}{3} F_2 s_3 = (1/3)(135)(7) = 315 \text{ in-lb}$$

The maximum bending moment at point A is

$$M_{max} = q_1 F_2 s_2 = (1.3)(135)(7.5) = 1316 \text{ in-lb}$$

and the minimum is

$$M_{min} = F_2 s_2 - F_1 s_1$$

or

$$M_{min} = 135(7.5) - (40.5)(8.25) = 678 \text{ in-lb}$$

The mean and range values of these loads are found from the foregoing to be

$$T_m = \frac{T_{max} + T_{min}}{2} = \frac{1229 + 315}{2} = 772 \text{ in-lb}$$

$$T_r = \frac{T_{max} - T_{min}}{2} = \frac{1229 - 315}{2} = 457 \text{ in-lb}$$

$$M_m = \frac{M_{max} + M_{min}}{2} = \frac{1316 + 678}{2} = 997 \text{ in-lb}$$

$$M_r = \frac{M_{max} - M_{min}}{2} = \frac{1316 - 678}{2} = 319 \text{ in-lb}$$

The maximum shear stress theory of failure for a hollow shaft [7] can be written as

$$\tau_{max} = \frac{.5 S_{yp}}{N} = B \left[K_{sb} \left(M_m + \frac{S_{yp}}{S_e} M_r \right)^2 + \left(T_m + \frac{S_{yp}}{S_{es}} T_r \right)^2 K_{st} \right]^{\frac{1}{2}}$$

where

$$B = \frac{16}{\pi D_o^3 \left\{ 1 - \left(D_i / D_{oB} \right)^4 \right\}}$$

Since the outer diameter D_o is constrained to a value of 1.375 inches, the equation is rearranged to solve for the maximum inner diameter:

$$D_i = \left[D_o^4 \left(1 - \frac{32 N A}{\pi D_o^3 S_{yp}} \right) \right]^{\frac{1}{4}}$$

where

$$A = \left[K_{sb} \left(M_m + \frac{S_{yp}}{S_e} M_r \right)^2 + \left(T_m + \frac{S_{syp}}{S_{es}} T_r \right)^2 K_{st} \right]^{\frac{1}{2}}$$

The values of the terms in these expressions are as follows:

$$S_e = C_R C_S C_F \frac{1}{K_f} S'_n$$

$$S_{es} = C_R C_S C_F \frac{1}{K_{fs}} S'_{ns}$$

$C_R = 1 - .08(\text{DMF}) = 0.81$ ----- Reliability for 99% survival rate

$C_S = 0.85$ ----- Size factor for bending and torsion

$C_F = 0.77$ ----- Finish factor for 1020 CD steel

$K_f = K_{fs} = 1.3$ ----- Fatigue stress concentration factor in bending and shear

$K_{sb} = K_{st} = 1.25$ ----- Minor shock loads

$N = 2$ ----- Factor of safety

For AISI 1020 CD steel the following values are used:

$$S_{yp} = 66,000 \text{ psi}$$

$$S_{syp} = .58 S_{yp} = 38,280 \text{ psi}$$

$$S'_n = .5 S_u = .5(78,000) = 39,000 \text{ psi}$$

$$S'_{sn} = .29 S_u = 22,620 \text{ psi}$$

$$S_e = 15,904 \text{ psi}$$

$$S_{es} = 9224 \text{ psi}$$

Substituting these values into the above equations produces

$$A = \left[1.25 \left(997 + \frac{66000}{15904} 319 \right)^2 + \left(772 + \frac{38280}{9224} 457 \right)^2 \right]^{\frac{1}{2}}$$

$$A = 3954 \text{ in-lb}$$

Thus

$$D_i = \left[1.375^4 \left(1 - \frac{32(2)(3954)}{\pi(1.375^3)(66000)} \right) \right]^{\frac{1}{4}} = 1.174 \text{ inches}$$

The fatigue analysis shows that a minimum tube thickness of 0.1 inches is required for a factor of safety of 2. However, because of the requirement of a 1/8 inch deep keyway, the specified tube thickness is 3/16 inches. This indicates an inner diameter of 1 inch.

Overload Analysis

Overload factor, $q_3 = 3$

Maximum vertical wheel load, $F_2 = 200 \text{ lb}$

Tire side force, $F_1 = 0.3(200) = 60 \text{ lb}$

Tire cornering force, $F_3 = 0.5(200) = 100 \text{ lb}$

Wheel offset to point A, $s_2 = 7.5 \text{ inches}$

Wheel radius, $s_1 = 8.25 \text{ inches}$

From these values, the maximum applied torsion T_{max} on the tube is 4200 in-lbs and the maximum bending moment results from the combination of the vertical wheel load, the tire side force and the maximum cornering force. The moment is given by

$$M_{max} = q_3 F_2 s_2 + F_1 s_1 + F_3 s_1$$

Substituting the values into the above equation produces a moment of 5820 in-lbs. From the maximum shear stress theory of failure

$$\tau_{max} = \frac{.5 S_{yp}}{N} = \frac{16}{\pi D_o^3 \left[1 - (D_i / D_o)^4 \right]} \left(M_{max}^2 + T_{max}^2 \right)^{1/2}$$

Using the specified inner diameter of 1 inch and solving indicates a factor of safety of 1.7.

APPENDIX D

TONGUE STRESS ANALYSIS

APPENDIX D

TONGUE STRESS ANALYSIS

The stress analysis of the tongue involves the determination of the maximum bending stress at the most critical location - the lower curved portion of the tongue shown. The tongue is welded to the underside of the trailer chassis and is assumed to be rigidly connected over the welded portion. Thus a cantilever effect is produced by the hitch load. With the assumed hitch load of 10% of the gross trailer weight, and for a 200 lb concentrated load as shown, the hitch load is 100 lb. Thus the applied moment at the region of interest is 2250 in-lbs and is defined positive since it tends to decrease the beam radius of curvature.

For an outer diameter of 1.5 inches, an inner diameter of 1.375 inches and a radius of curvature R of 3 inches, the shape factor for the hollow tube [9] is

$$z = -1 + \frac{2R}{c_2^2 - c_1^2} \left[\left(R^2 - c_1^2 \right)^{1/2} - \left(R^2 - c_2^2 \right)^{1/2} \right]$$

where

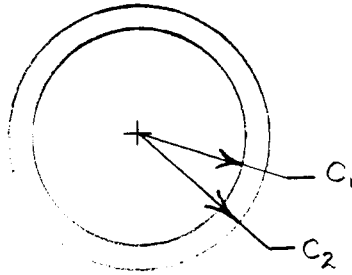
$$c_1 = \frac{1.375}{2} = 0.6875 \text{ in}$$

$$c_2 = \frac{1.5}{2} = 0.75 \text{ in}$$

$$A = \frac{\pi}{4} (D_o^2 - D_i^2) = 0.2823 \text{ in}^2$$

Thus

$$z = 0.30061$$



Tube cross-section parameters.

The Winkler-Bach expression for stress [9] is

$$\sigma = \frac{M}{AR} \left[1 + \frac{y}{z(R+y)} \right] + \frac{P}{A}$$

where y assumes the value $y_c = -0.75$ corresponding to the region of higher stress on the curved beam. Substituting values into the equation yields

$$\sigma = \frac{2250}{(.2823)(3)} \left[1 - \frac{.75}{.030061(3 - .75)} \right] + \frac{100 \sin 60^\circ}{.2823} = -26,496 \text{ psi}$$

For AISI 1010 CD steel, the yield strength is 68,000 psi. This analysis indicates a factor of safety of 2.5.

APPENDIX E

COST ANALYSIS

APPENDIX E

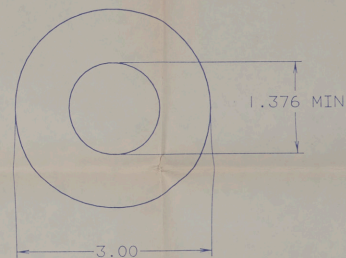
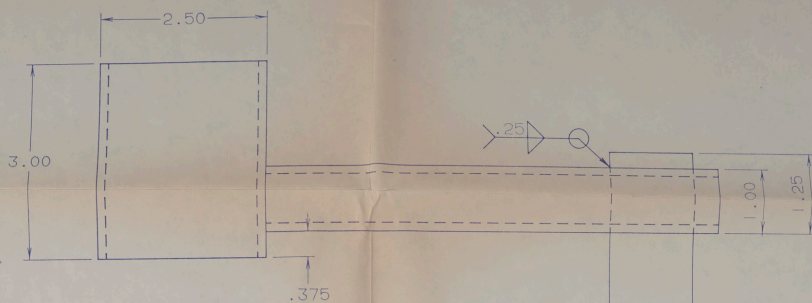
COST ANALYSIS

The cost analysis is based on a total of 5000 units. The labor costs reflect U.S. labor cost at the time of development and are divided into 3 levels: skilled, semi-skilled and unskilled. A breakdown of the cost is presented in the table below.

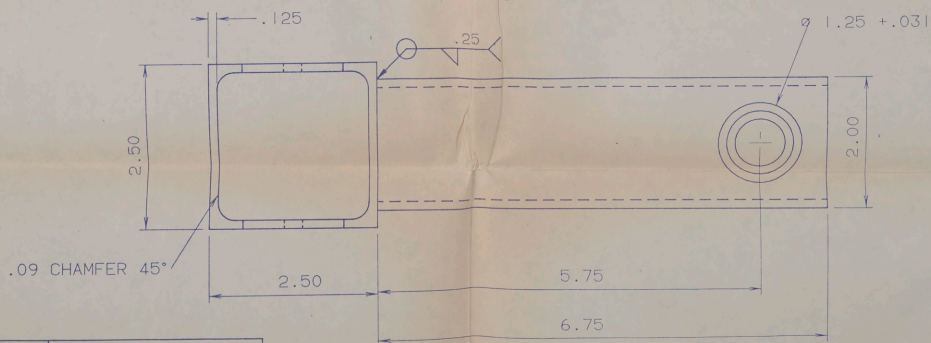
PART	MATERIAL COST	LABOR COST	QUANTITY	COST PER TRAILER
Rubber spring	\$11.71	—	2	\$23.42
Wheel / Tire	\$13.74	—	2	\$27.48
Hub / Bearing	\$12.58	—	2	\$25.16
Spindle	\$3.00	—	2	\$6.00
Trailing arm	\$1.00	\$3.43	2	\$8.86
Chassis plate	\$2.20	\$3.20	1	\$5.40
Cross tube	\$6.00	\$6.00	1	\$12.00
Tongue	\$4.50	\$3.00	1	\$7.50
Coupler	\$1.50	\$6.00	1	\$7.50
Trailer bed	\$4.00	\$2.00	1	\$6.00
Hardware	\$1.50	—	1	\$1.50
				<u>\$130.82</u>

VITA

Olakunle Harrison was born in Ibadan, Nigeria on June 20, 1959. He attended high school in Nigeria and began his college education at Carson Newman College, Jefferson City, Tennessee. He transferred to the University of Tennessee in January, 1984. He was graduated with a Bachelor of Science degree in Mechanical Engineering in March, 1986 and began graduate school in the same year. His major interests are automotive mechanics and mechanical devices.

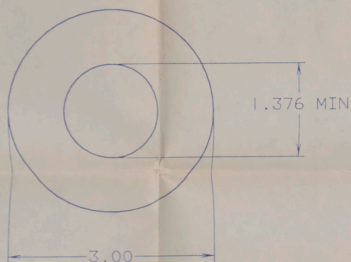


FLAT WASHER (4), STEEL

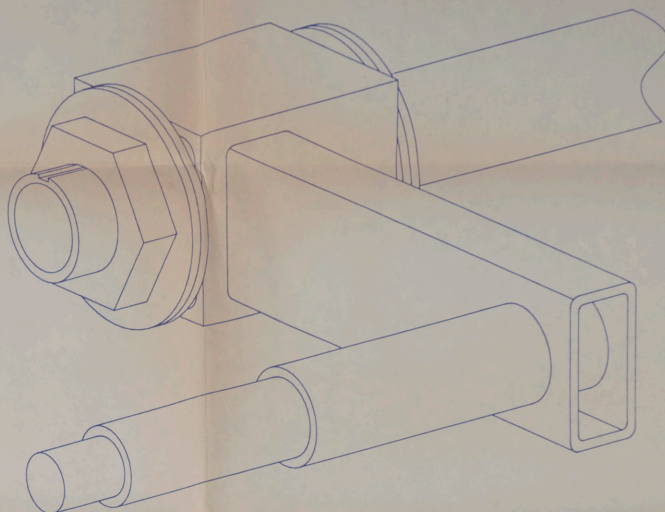
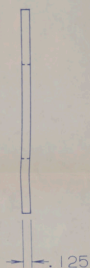


UNITS	INCHES
MAT'L	STEEL
TOLERANCES	UNLESS NOTED
LINEAR±	.005
ANGLE±	1°
SURFACE TREATMENT	NONE

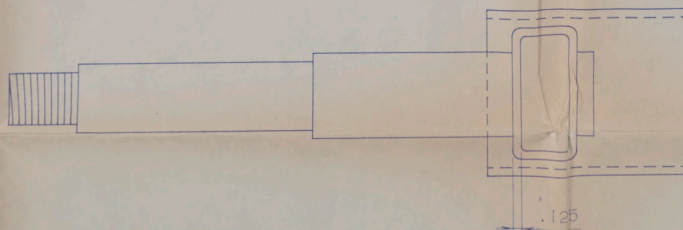
TRAILING ARM
MAT'L STEEL, 2 REQ'D



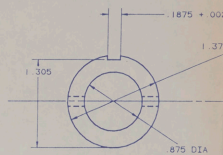
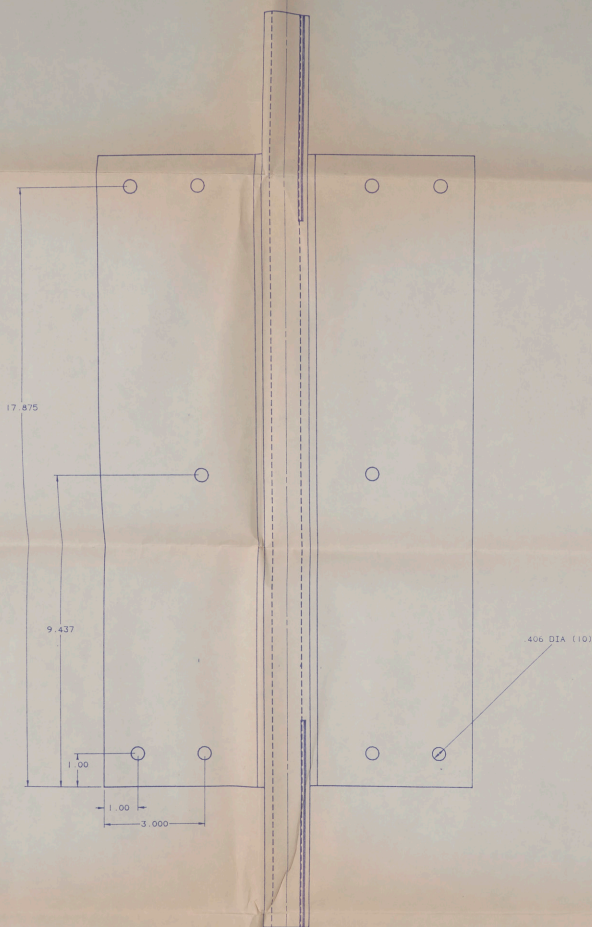
FLAT WASHER (4), STEEL



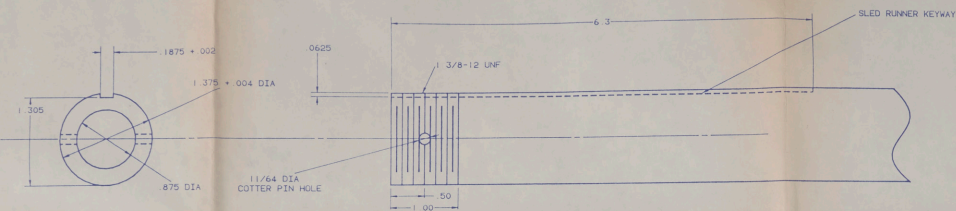
ASSEMBLY ON SHAFT



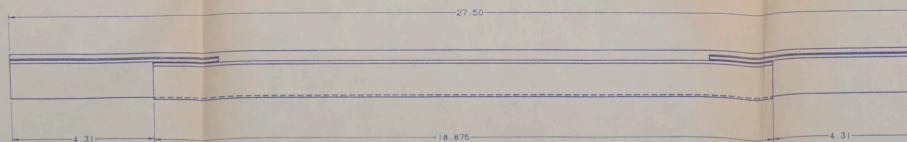
UNIVERSITY OF TENNESSEE ENGINEERING COMPUTING CENTER	
TITLE	TRAILING ARM
DESIGNER	OLAKUNLE HARRISON
SECTION	PLATE 1
SCALE	FULL
DATE	JULY 22, 1988



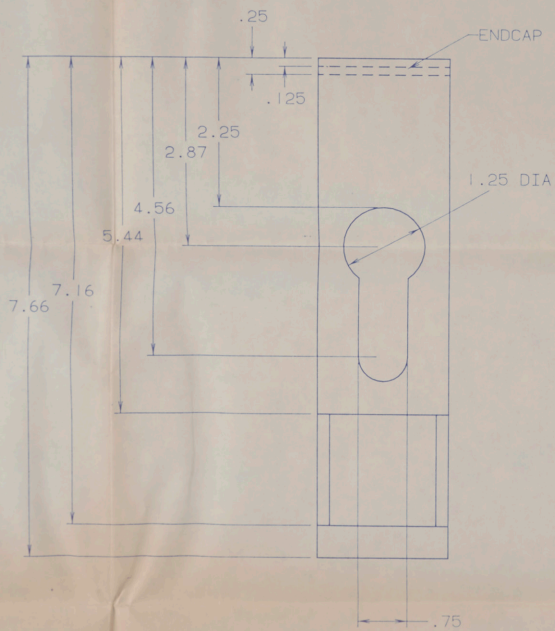
UNITS	INCHES
MAT'L	STEEL
TOLERANCES UNLESS NOTED	
LINEAR ±	.005
ANGLE ±	1°
SURFACE TREATMENT	NONE



SHAFT DETAILS

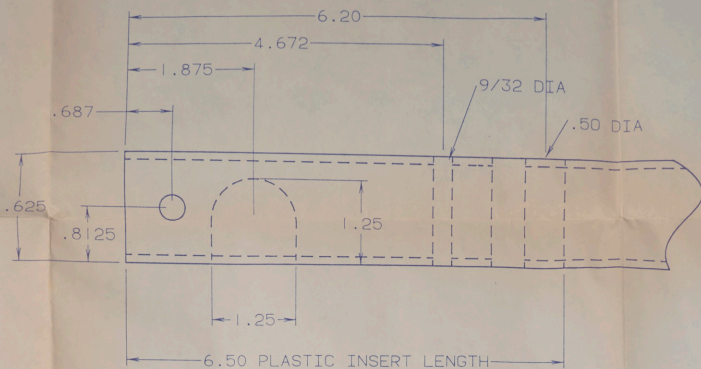


UNIVERSITY OF TENNESSEE ENGINEERING COMPUTING CENTER	
TITLE	CHASSIS PLATE
DESIGNER	OLAKUNLE HARRISON
SECTION	PLATE 2
SCALE	1:2
DATE	AUGUST 15, 1988



COUPLER SLEEVE

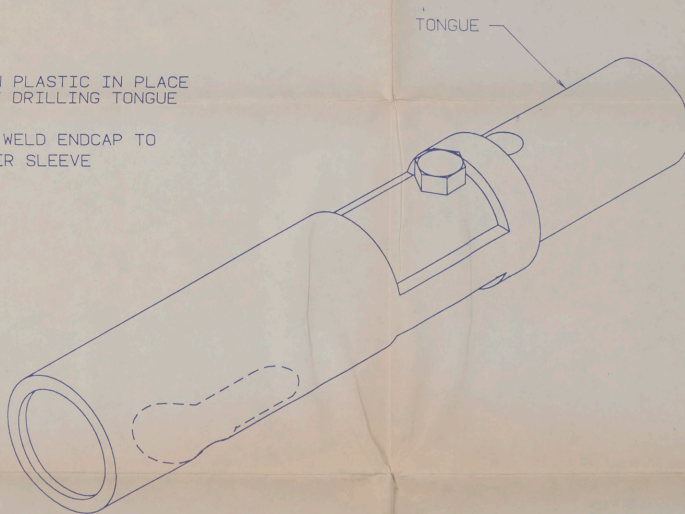
UNITS	INCHES
MAT'L	STEEL
TOLERANCES UNLESS NOTED	
LINEAR±	.005
ANGLE±	1°
SURFACE TREATMENT	NONE



TRAILER TONGUE/COUPLER

NOTES:

1. DELRIN PLASTIC IN PLACE BEFORE DRILLING TONGUE
2. FILET WELD ENDCAP TO COUPLER SLEEVE



COUPLER ASSEMBLY

UNIVERSITY OF TENNESSEE ENGINEERING COMPUTING CENTER	
TITLE	TONGUE/COUPLER
DESIGNER	OLAKUNLE HARRISON
SECTION	PLATE 3
SCALE	1:1
DATE	MARCH 12, 1988