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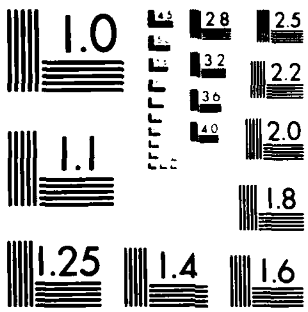

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SPREADER BAR

ALIGNMENT ARM DEVELOPMENT STUDY

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VSE CORPORATION
2550 Huntington Avenue
Alexandria, Virginia 22303

April 1983
Approved for public release; distribution unlimited

Final Report

Prepared for:

Mechanical and Construction Equipment Laboratory
U.S. Army Mobility Equipment
Research and Development Command
Fort Belvoir, Virginia 22060

NOTICE

SPREADER BAR

ALIGNMENT ARM DEVELOPMENT STUDY

Prepared for:

U. S. Army Mobility Equipment Research and
Development Command
Fort Belvoir, Virginia 22060

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Ralph J. Adolph
Central Engineering Division

This technical report was written in accordance with the requirements of MIL-STD-847A.

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REPORT DOCUMENTATION PAGE

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

This report documents development of energy damping alignment arms for solution of dynamic alignment problems occurring during freight operations at sea using spreader bars for lifting ISO and Freight Containers.

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SUMMARY

This report documents development of energy damping alignment arms for solution of dynamic alignment problems occurring during freight operations at sea using spreader bars for lifting ISO and freight containers.

This report considers solutions to the dynamic alignment problem in positioning the spreader bar for lifting ISO and freight containers. Both passive and active systems were considered. Three passive system concepts were evaluated, with conclusions and recommendation for implementation of one of the concepts. Additionally, the report documents a preliminary force study conducted to evaluate the prevalent forces due to the high energy and momentum states of the spreader bar/container system.

SPREADER BAR

ALIGNMENT ARM DEVELOPMENT STUDY

TABLE OF CONTENTS

1.0 BACKGROUND

Spreader bars are utilized by the military and by commercial firms to facilitate cargo transfer operations involving standardized ISO containers. Cranes with attached spreader bars are used to transfer containers from cargo ships to docks and vice versa in normal freight operations. During the operations, the crane operator maneuvers the spreader bar, with the help of deck or ground crews using tag lines, such that it is vertically aligned with the container top. Bayonet studs in the four corners of the spreader are engaged in fittings at the corners of the container. The ground or deck crews operate cables on the spreader bar which lock the bayonet studs into the container fittings. The crane operator then lifts the container and transfers it. The ground or deck crews then disengage the bayonet studs and the crane operator lifts the spreader bar and positions it over the next container to be transferred.

1.1 Statement of Technical Problem

During military operations, these freight transfer operations frequently are conducted under less than ideal conditions. When the operations are conducted in shallow harbors, the containers can be transferred at sea from the cargo ship to barges, then from the barges to land. Because it is tactically unfeasible to restrict these operations to periods of calm sea states, a requirement was established for freight operations during sea state 3. The relative motion between the transferal equipment at sea state 3 necessitates the use of energy-absorbing mechanisms to facilitate rapid transfer. Consequently, a program was initated to develop energy-absorbing alignment arms for the spreader bars. The first phase of the program was intended to identify energy levels and the forces that exist between the spreader bars and containers. The second phase involved establishing feasible alternative concepts and the third phase was intended to document the selected design.

1.2 Spreader Bar Force Analysis

A preliminary force analysis was accomplished and is provided in appendix A. The purpose was to examine the energy and momentum states of the spreader bar for sea state 3. The energy and momentum states were used to establish the equations governing the physical requirements for alignment arms such that an analysis of alternative designs could be conducted. The intent was that the forces created during transer operations, with any of the design concepts, would not deteriorate the structural integrity of either the spreader bar or the container.

It was concluded that it would not be feasible to attempt to absorb the vertical energy existing between the spreader bar and the container. The energy levels are too high and the vertical displacement necessary for the energy absorption is not available. Absorption of all of the lateral energy in one pass of the spreader bar necessitates approximately three feet of travel for the alignment arm.

1.3 Evaluation of Alternative Concepts

The evaluation of alternative concepts for solving the dynamic alignment problem for engaging a spreader bar with an ISO container was approached by considering concepts encompassing a broad scope. Appendix B is a report summarizing the concepts and evaluation methodology. Approaching the problem's solution systematically, it was determined that active or passive systems were technically feasible. The active-system approach required that the crane operator be considered part of the system. The operator would receive feedback with respect to the dynamic state and relative position of the spreader bar and container and modify the state and position through system controls. The passive-system approach eliminated or minimized the crane operator's participation beyond lowering the spreader bar and centering its motion over the container.

The passive-system concepts were of two types. The first utilized rigid alignment arms attached to the spreader bar through energy-absorbing mechanisms. The second utilized wire-rope cables directly affixed to the container with a lightweight frame by the deck crew and indirectly attached to the spreader bar so that it is drawn down the cables to the container.

The evaluation of the concepts involved an informal trade-off study of the following features:

- Capability to perform primary mission
- Capability for positioning of articulating alignment arms between stowed and ready positions
- Capability of withstanding backside impacts
- Transmission of vertical forces to the container top
- Capability of withstanding side impacts
- Relative operational complexity
- Relative mechanical complexity
- Exposure of functional mechanisms to elements
- Structural integrity
- Estimated fabrication costs

A recommendation was made for implementing the design that was improved and is described in detail in section 2 and assessed in section 3.

2.0 DESIGN DESCRIPTION

The design that is described herein and in the documentation package is intended for use on a 20-foot spreader bar. Minor mounting modifications would extend its use to 35- or 40-foot spreader bars.

2.1 Overview

Six alignment arms are attached to a 20-foot spreader bar as depicted in figure 1. Details are shown in figure 2. Operationally, three devices are placed in the ready position while three remain in the stowed position (two on one side and one on an end). The crane operator swings the spreader bar in over the container at an angle such that the three devices engage the container when the spreader bar is directly overhead. As the spreader

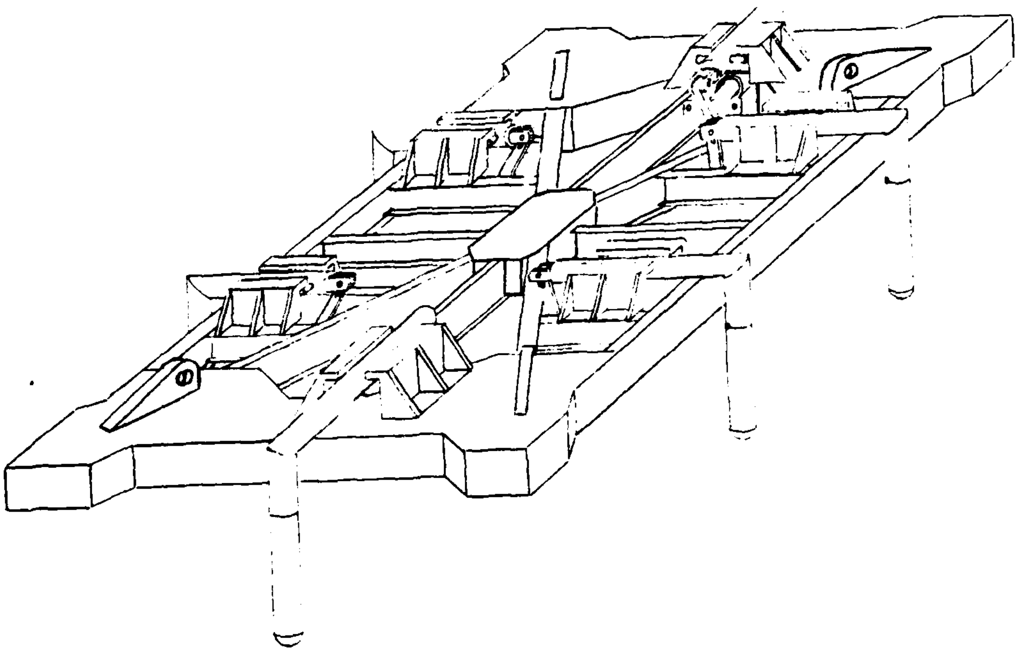


Figure 1. Spreader Bar with Energy Dampening Alignment Arms

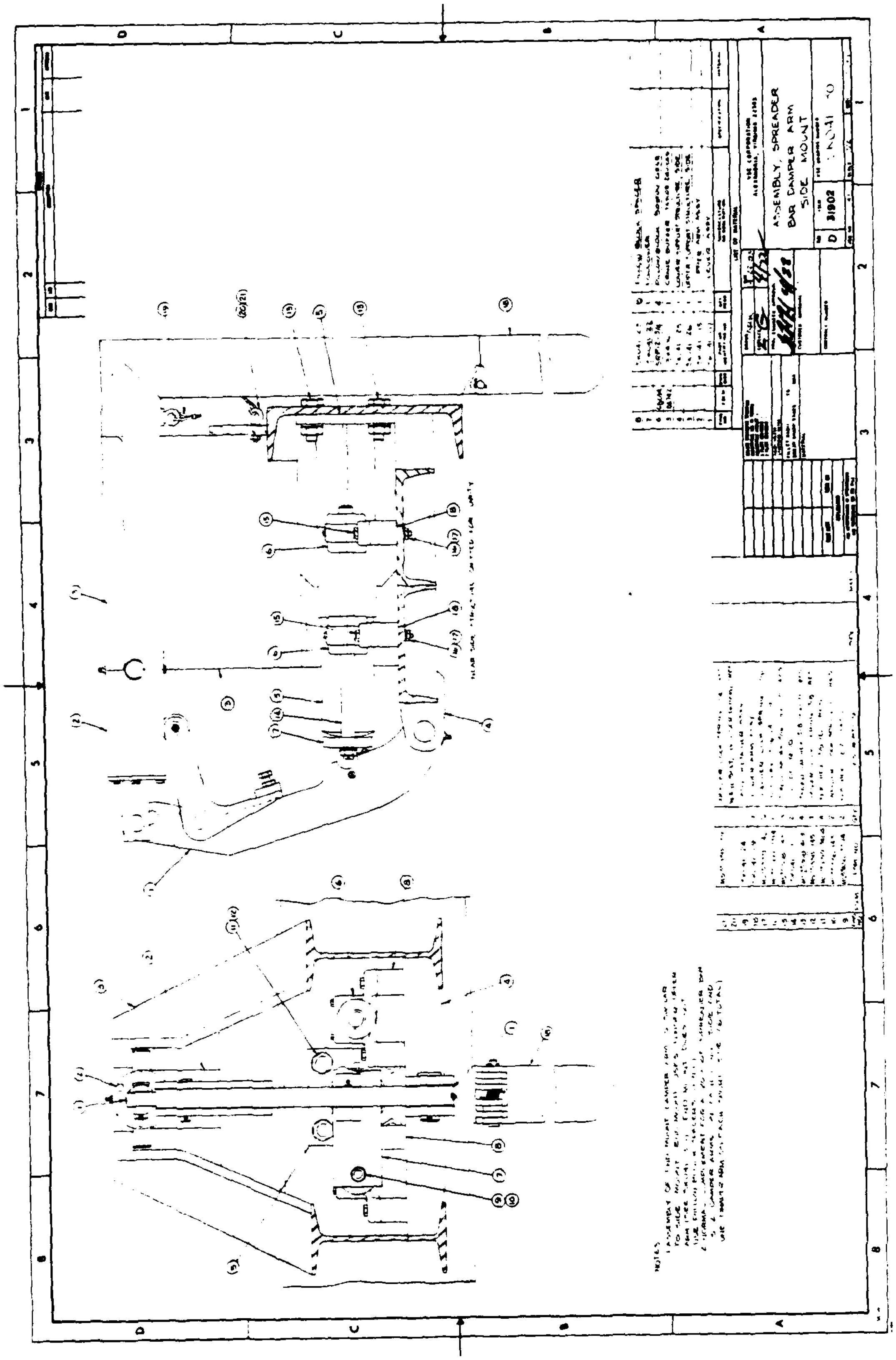


Figure 2. Energy Damped Alignment Arm Detail

bar continues its motion, the three alignment arms extend and thus load their respective shock absorbers. If the kinetic energy is absorbed within the stroke of the mechanisms, the return springs in the shock absorbers reverse the spreader bars' motion and bring it to rest centered over the container. Should the energy level of the spreader bar be too high, the alignment arms will release and swing up, allowing the spreader bar to pass over the container at a reduced energy state. The alignment arms are heavy-wall, high-strength steel tubes attached to a lever pivoted at the base of the spreader bar. The lever has an imbedded roller that abuts a plate constrained to horizontal motion through rods that translate through sintered bronze bushings. The plate abuts a self-adjusting, oil-filled shock absorber mounted to either the side wall or the end wall of the spreader bar.

As the alignment arm engages the container, a moment tends to rotate it about the pin at the top of the lever. This moment is resis by a spindle or roller in the alignment arm. The roller is a shaft ex ding out of both sides of the alignment arm. The two extensions of the rol engage tracks or hold-down flanges resisting the vertical motion cause by the moment. As the roller travels along the track, the lever actua the shock absorber. As the shock absorber approaches its full stroke, the roller disengages from the track allowing the alignment arm to pivot at the upper lever pin and to swing up, disengaging the container. The swinging motion of the alignment arm is arrested by a wire-rope cable as the return spring is resetting the shock absorber. The alignment arm then swings down until the roller extensions engage rubber pads atop the tracks. The return spring resets the alignment arm with the roller back in its original position in the track.

2.2 Description of Key Components

2.2.1 Shock Absorber

The shock absorber is a heavy-duty industrial model that absorbs kinetic energy, converting it to heat. During its stroke, oil is forced through an orifice. The compression return spring is completely internal. The unit has Teflon seals and can operate in an environment from -40^{o}F to +160^{o}F ambient. The resisting force varies with impact velocity. The unit is protected from salt-spray environment by a rubber bellows enclosure.

2.2.2 Lever

As previously mentioned, the lever is pivoted at the base of the spreader bar and has an imbedded roller. It provides a mechanical advantage of approximately 3.7 to 1. The upper attachment point is slotted to facilitate engagement and disengagement of the alignment arm spindle in the track. When the alignment arm encounters vertical forces from the container top, the pin attaching it to the lever can travel down in the slot allowing the spindle to back out of the track causing the alignment arm to swing up clear of the container. An adjustment arm is attached to the lever. This adjustment arm limits the angular orientation of the alignment arm with respect to the lever and retains the attachment pin in the top of the slot when the mechanism is in the ready position.

As the alignment arm travels through the track, the attachment pin first travels down and then back up in the slot. The adjustment arm's position is varied at installation to provide proper engagement of the alignment arm in the track. A jam nut secures the adjustment arm at its proper setting.

2.2.3 Clevis/Spindle Assembly

The alignment arm is attached to the lever through a rotational clevis. One end of the clevis is attached to the lever while the other end is captured in the alignment arm. The alignment arm is free to rotate about the clevis when its spindle is not engaged in the track. When in the ready position, if the alignment arm is impacted directly from the side, the slotted lever allows the spindle to back out of the track and the alignment arm will rotate about the clevis swinging free from the impact source.

The spindle or roller is captured in the alignment arm by an internal collar. The collar is pinned to the spindle with a roll pin.

2.2.4 Hold-Down Flange

The hold-down flange provides a track for the alignment arm's spindle. In the side-mount configuration it is attached to the spreader bar through one existing I beam and another added I beam. It is gussetted to provide adequate stiffness. Rubber pads are cemented to the flange top to avoid damage to the spindles. In the end-mount configuration it is attached to the top plate of the spreader bar.

2.2.5 Hinged Alignment Arm

The alignment arm is hinged at the base of the spreader bar. It is a torsion spring-loaded return hinge and allows the lower portion of the alignment arm to swing under the spreader bar when it is impacted from the backside. A hemispherical end cap is welded to the bottom of the alignment arm to eliminate sharp corners that could damage a container top.

3.0 RISK ASSESSMENT

As with any untested design, there are several risks associated with the alignment arm design that should be identified, although their impact is deemed to be minimal with respect to the overall functionality of the design.

3.1 Shock Absorber Capacity

The nature of the design is such that the shock absorber will not be loaded beyond its full stroke. The device will be orificed such that it will produce a force that will not be detrimental to the components at the maximum expected relative velocities. If, however, the relative velocity exceeds the designated maximum during an operation, the seals within the shock absorber can be damaged. The unit can be disassembled for repair when taken out of service. An alternative would be to specify a nonstandard unit that would have an internal pressure relief when the orifice capacity is exceeded. At this point the added expense is not considered necessary.

3.2 Hold-Down Flange Engagement

Operational conditions might occur that could exert forces on the alignment arm when it is not completely engaged in the track. This would result in the alignment arm swinging free when it should be engaged. Other complications can also arise, e.g., an oblique side loading keeping one extension of the spindle in the track while forcing the other extension down. The slot in the lever and lever bending within the elastic limit should prevent damage but testing would be required to establish a limiting condition. It also might be required to give a more generous acceptance angle at the entrance to the track to ensure proper engagement.

3.3 Trapped Hinged Alignment Arm

When impacted from the backside, the lower portion of the alignment arm is hinged such that it will swing under the spreader bar. The possibility exists that the arm could be trapped between the spreader bar and container. It could be freed by raising the spreader bar but if the crane operator is not careful, the arm could wedge itself and cause damage to the container top.

3.4 Corrosion and Wear

Corrosion is a problem that has been given careful consideration throughout the design of the system. Lubrication points have been provided for all of the moving parts. The frequency of lubrication has yet to be estabished, however.

Wear of components that are subjected to impact loading has also been considered. The adjustable arm for the lever is one of the design improvements that was implemented due to wear considerations. If the device was being designed for commercial use requiring continual operations, other improvements might be considered. For tactical military use, however, wear should not have a major impact.

4.0 CONCLUSIONS

Evaluation of the dynamic conditions expected during freight transfer operations at sea, consideration of alternative concepts, and detailed evaluation of the design lead to the conclusion that the system proposed here has a high probability for successful implementation.

5.0 RECOMMENDATIONS

It is recommended that the system be fabricated with a limited testing program conducted at the subassembly level and extensive operational testing conducted for the overall system.

APPEXDIX A

SPREADER BAR
FORCE ANALYSIS

Table of Contents

Technical Report

Spreader Bar Force Analysis

1.0 Background

Spreader bars are structurally strong metal devices, of three different dimensions to be compatible with 20, 35, and 40 feet containers. The spreader bar, suspended from a crane at the four lift lug attachments, is lowered onto a container top, where four bayonet studs on the spreader bar are inserted simultaneously into the four bayonet-accepting holes in the container top (see figure 1).

Spreader bars are utilized to facilitate transfer of military ISO containers during freight loading and unloading operations. When it is necessary to perform these operations at sea, dynamic alignment problems can occur even during relatively mild sea state conditions. While these problems are a concern for standard commercial operations, during military operations where rapid transfer is a necessity, the need to solve the dynamic alignment problem is urgent.

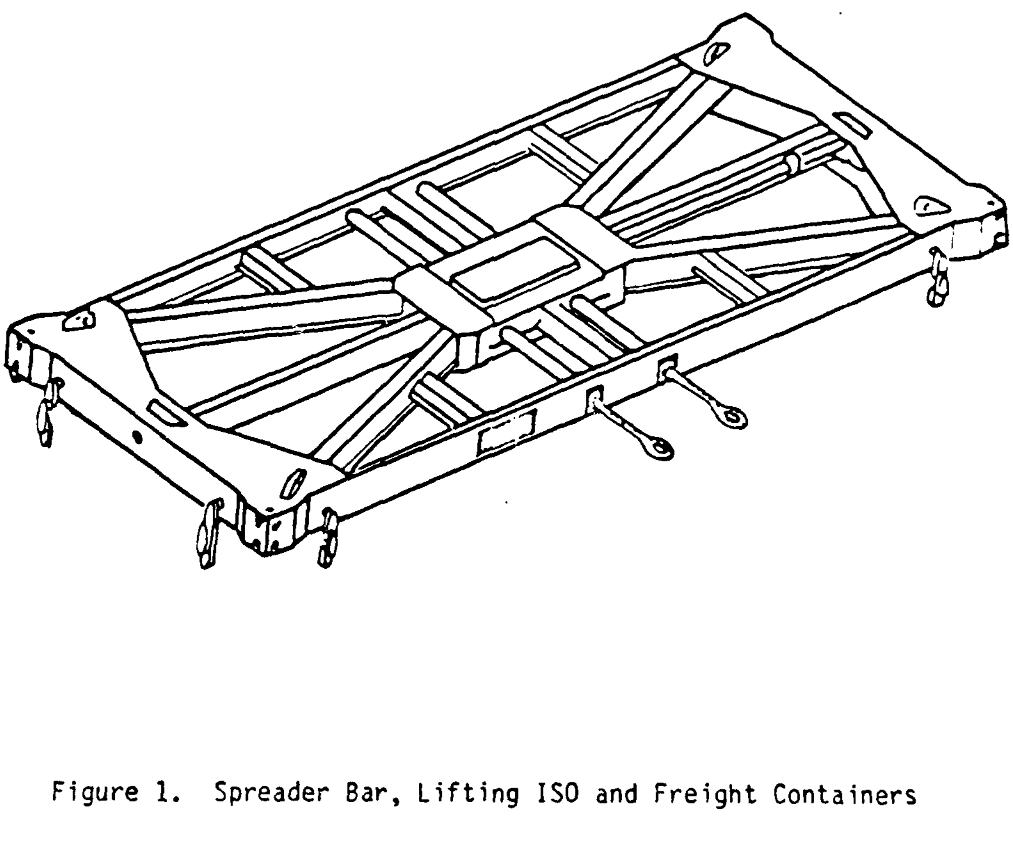


Figure 1. Spreader Bar, Lifting ISO and Freight Containers

One of the approaches to the problem's solution is to attach energy absorbing alignment arms to the spreader bar.

2.0 Purpose

The purpose of this report is to examine the energy and momentum states of the spreader bar for a set sea state condition. The information is to be used to establish the equations governing the physical requirements for alignment arms such that analysis of alternative designs can be conducted and such that forces created during transfer operations utilizing the alignment arms will not deteriorate the structural integrity of either the spreader bar or the container.

3.0 Technical Analysis

The technical approach to the analysis consisted of the following:

a. The government furnished information with respect to weights of three spreader bar sizes and relative velocities between the spreader bar and container. These relative velocities represent the governments best estimate of conditions present during the worst sea states existing during transfer operations. Based on the velocities and masses received the momentum and kinetic energy of the largest spreader bars were calculated. Assuming the spreader bar could be brought to rest in a one foot horizontal distance, the constant horizontal force required would be 18,221 lbf. Similarly, if the spreader bar could be brought to rest in .5 feet of vertical motion, the constant force required would be 10,285 lbf. In order to determine the size of an alignment arm to accommodate these force levels, an alignment arm configuration was assumed (see figure 2). The minimum diameter was determined to be 3.5 inches. These computations may be found at Appendix B.

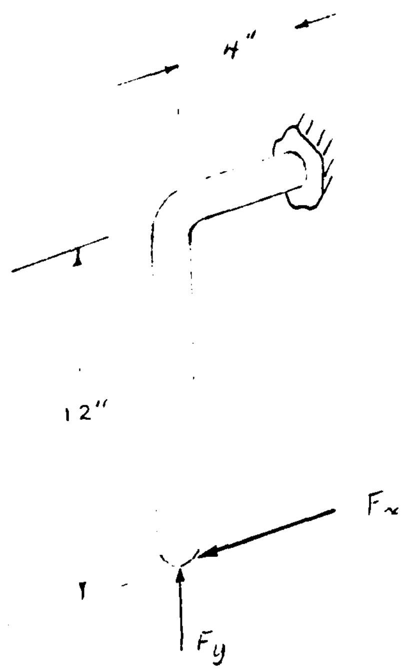


Figure 2. Alignment Arm Force Diagram

b. The ISO container was then evaluated. Experience indicates that the top is the most susceptible to damage. Therefore, the force levels and the alignment arm established in Appendix B were used to establish the stress levels on the top of the container. The top shown in figure 3 would have a stress level of 148,000 psi in resisting the vertical motion of the spreader-bar. This would be a stress level in the steel perimeter, if the alignment arm struck the aluminum center the stress levels would be much higher. These calculations are at Appendix C.

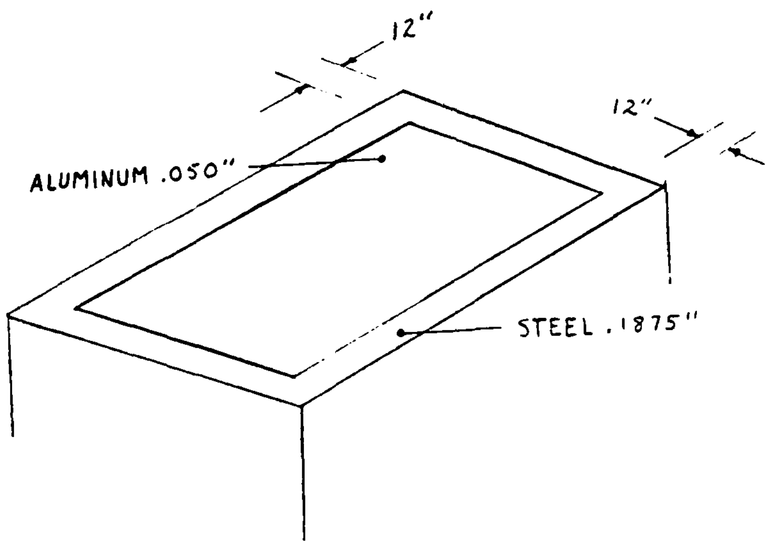


Figure 3. ISO Container Top

c. Assuming an energy absorbing alignment arm, the physical components required to accomplish this can be modeled as an idealized spring/damper combination. The idealized damping and spring constants were computed assuming critical damping. Parametric (time) force and displacement equations were developed, again assuming the spreader bar's motion would be stopped in one foot. The results are shown in table 1 and the calculations are at Appendix D. Included in the table are the cumulative increments for the impulse on the alignment arm and work done by the alignment arm. This table was developed to compare the impulse and work to the initial momentum and kinetic energy determined for the spreader bar. This table shows the variation of force for the idealized mechanical system. Thus the constant force estimate gives a much lower force than can be realistically expected. This indicates that arresting the motion of the spreader bar in a one foot is unrelealistic. Possibly two to three feet would be required to attain reasonable force levels.

Table 1. Force Distance Time Relationship
During Idealized Spring Damper Deceleration

Time Sec t	Distance Ft x	Force lb F	Impulse lb-s $\Sigma F\Delta t$	Work lb ft $\Sigma F\Delta x$
0	0	26,900	26.9	431.0
.001	.016	26,820	53.7	849.0
.002	.0316	26,400	80.1	1287.0
.003	.0472	26,200	184.8	2858.0
.007	.1073	25,250	260.5	3958.9
.010	.1509	24,600	383.5	5646.5
.015	.2195	23,500	735.5	9935.2
.030	.4020	20,500	1349.5	15470.2
.060	.6720	15,500	1659.5	17392.2
.080	.7960	12,750	1914.5	18526.9
.100	.8850	10,450	2123.5	19153.9
.120	.9450	8,500	2378.5	19519.4
.150	.9880	6,760	2716.5	19458.6
.200	.9790	3,325	2550.3	19235.8
.250	.9120	1,571	2413.5	19039.4
.337	.7870	0		

Where:

$$Imp_x = Fdt \doteq \sum F_i \Delta t_i$$

$$KE_x = Fdx \doteq \sum F_i \Delta x_i$$

The figures for impulse and work (utilizing a course integration method) approximate the momentum and kinetic energy, respectively, of the spreader bar calculated in Appendix B indicating that the solution is correct.

4.0 Conclusions and Recommendations:

- Horizontally the energy absorbsion must take place over a relatively long time period, that is, several passes of the spreader bar over the container will be required to reduce the dynamic alignment problem to a level that can be managed with tag lines.

- The alignment arms should be "tuned" for the heaviest spreader bar. *While not optimized for the lighter spreader bars, the force levels* will be lower due to the decreased mass.

- The alignment arm should not be capable of transmitting vertical forces of any appreciable magnitude. It would be difficult to design an adequate vertical energy absorbing alignment arm that would not deteriorate the steel perimeter of the container top. The aluminum center of the top can withstand much lower force levels than the perimeter.

APPENDIX A

References

a. Velocities and masses of the spreader bar, during operations in the worst case sea state expected and the configuration of the container top were provided during meetings and telephone conversations with Steve Goguen, MERADCOM, during August 1982.

b. Thomson, W.T., "Vibration Theory and Applications", Prentice-Hall, 1965.

c. Juvinall, R.C., "Stress, Strain, and Strength", McGraw-Hill, 1967.

d. Roark, R.J., "Formulas for Stress and Strain", McGraw-Hill, 1965.

APPENDIX B

Determination of Spreader Bar Dynamic Conditions

Spreader bar weights (Government furnished information)

20' - 2,625 lbs.
35' - 4,175 lbs.
40' - 4,580 lbs.

Assumed relative velocity between spreader bar and container at contact (Government furnished information)

Horizontal $V_x = 16$ ft/s

Vertical $V_y = 8.5$ ft/s

Momentum for 40' spreader bar

$$Imp_x = M_{40'} V_x$$
$$= (4,580)\ (16)\ \frac{1}{32.174}$$

$$Imp_x = 2,278 \text{ lbf-s}$$

$$Imp_y = M_{40'}\ V_y$$
$$= (4,580)\ (8.5)\ \frac{1}{32.174}$$

$$Imp_y = 1,210 \text{ lbf-s}$$

Kinetic Energy For 40' Spreader Bar

$$KE_x = \tfrac{1}{2} M_{40'}\ V_x^{\ 2}$$
$$= \tfrac{1}{2}\ (4,580)\ (16)^2\ \frac{1}{32.174}$$

$$KE_x = 18,221 \text{ ft lbf}$$

$$KE_y = \tfrac{1}{2} M_{40'}\ V_y^{\ 2}$$
$$= \tfrac{1}{2}\ (4,580)\ (8.5)^2\ \frac{1}{32.174}$$

$$KE_y = 5,152.4 \text{ ft lbf}$$

Impulse Momentum

$$\int F_x dt = Imp_x$$

$$= \int mdv_x$$

$$F_x \cdot = m \frac{dv_x}{dt}$$

$$= m \frac{dv_x}{dx} \quad \frac{dx}{dt}$$

$$F_x = mV_x \frac{dV_x}{dx}$$

Assuming constant deceleration for one foot to zero velocity

$$\int V_x dt = \int dx$$

$$\int atdt = 1 \text{ ft}$$

$$\frac{1}{2} at^2 = 1 \text{ ft}$$

$$a = \frac{dV}{dt}$$

$$at = 16 \text{ ft/s}$$

$$\frac{1}{2} at^2 = 1 \text{ ft}$$

$$\frac{1}{2} t = \frac{1}{16} \text{ s}$$

$$t = .125 \text{ s}$$

$$a = 128 \text{ ft/s}^2$$

$$F_x = (4,580)\ (128)\ \frac{1}{32.174}$$

F_x = 18,221 lbf (This assumes constant deceleration. The force level due to a tuned mechanical system is examined in Appendix D.)

Similarly for 0.5 ft of vertical motion

$$F_y = 10{,}284.8 \text{ lbf}$$

Assuming a 1 foot cantilevered alignment arm x 4" takes the entire force

$$\sigma = \frac{Mc}{I}$$

$$\sigma = \frac{(F_x)\ (1) + (F_y)\ (\frac{4}{12})\ \ C}{I}$$

Letting σ max = 60,000 psi

$$\frac{I}{C} = \frac{((18{,}221) + (10{,}284.8)\ (\frac{1}{3}))\ 12}{60{,}000}$$

$$\frac{I}{C} = 4.3298 \text{ in}^3$$

For a cylindrical rod

$$C = \frac{Dia}{2}$$

$$I = \frac{\pi}{64}\ (Dia)^4$$

$$\frac{\pi}{32}\ (Dia)^3 = 4.3298 \text{ in}^3$$

$$Dia = \left(\frac{(32)\ (4.3298)}{\pi}\right)^{1/3}$$

$$Dia = 3.5331 \text{ in}$$

APPENDIX C

Container Stress Analysis

Contact Stress in Container Top

Radius of Circular Contact Area from Roark

$$r_o = 0.881 \left(\frac{PD}{E}\right)^{1/3}$$

$$= 0.881 \left(\frac{((18,221)^2 + (10,285)^2)^{1/2}(3.5331)}{30\times10^6}\right)^{1/3}$$

$$r_o = .119 \text{ in}$$

$\frac{3}{16}$ th in steel) Steel Perimeter
50,000 psi yield)

.050 Alum) Aluminum Center
35,000 psi yield)

If the steel plate (3/16 th) is supported adequately it can be treated as a rectangular plate 12" x 24" loaded at its center with a uniform load over a small circle of radius "a" with all edges supported. The top frame is constructed to provide this support).

$$\text{Max } s = \frac{3W}{2 \ mt}\left((m+1)\log\frac{2b}{r_o} + 1 - m\right)$$

$$\text{Max } y = \frac{Wb^2}{Et^3}$$

(from Roark)
s - stress
y - deflection
and are tabulated in Roark

W = total applied load

m = (reciprocal of Poisson's ratio)

t = plate thickness

b = plate width

a = plate length

r_o = radius of uniform load

W = 10,284.8 lbf

m = 3.3333

t = .1875 in

b = 12 in

a = 24 in

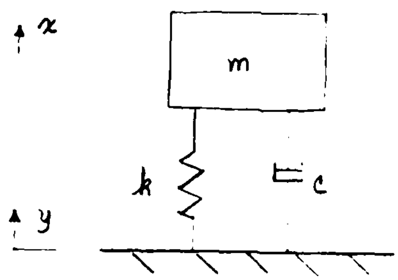
r_o = .119 in

$$\text{Max } s = \frac{3\ (10{,}284.8)}{2\ \ (3.3333)\ (.1875)}\left((3.3333+1)\ \log \frac{2\ (12)}{(.119)} + 1 - (.042)(3.333)\right)$$

$$= 148{,}459 \text{ psi}$$

APPENDIX D

Alignment Arm Requirements



Spring/Damper Combination

$$m\ddot{x} + c\dot{x} + kx = c\dot{y} + ky$$

Spring Only

$$m\ddot{x} + kx = ky$$

Damper Only

$$m\ddot{x} + c\dot{x} = c\dot{y}$$

The solution to the spring/damper eqn

$$x = Ae^{S_1 t} + Be^{S_2 t}$$

$$S_{1,2} = -\frac{C}{2M} \pm \left(\left(\frac{C}{2M}\right)^2 - \frac{k}{m}\right)^{\frac{1}{2}}$$

$$\omega_n^2 = \frac{k}{m}$$

and critical damping $C_c = 2(km)^{\frac{1}{2}} = 2m\omega_n$

ζ - Critical Damping Ratio

$$\zeta = \frac{C}{C_c}$$

$$S_{1,2} = -\zeta \pm (\zeta^2 - 1) \quad {}_n$$

Critical damping gives the minimum time to halt motion without oscillation.

For the critically damped case

$$x = (A + Bt)e^{-\omega_n t}$$

$$x = (x_o + (\frac{v_o}{\omega_n} + x_o)\,\omega_n t \quad e^{-\omega_n t}$$

with initial displacement x_0

and initial velocity v_0

of mass m

Given an initial velocity of 16 ft/s can the spreader bar be stopped in a reasonable x

$$\dot{x} = \left(\frac{v_0}{\omega_n} + x_0\right)\omega_n \; e^{-\omega_n t} - \omega_n \left(x_0 + \left(\frac{v_0}{\omega_n} + x_0\right)\omega_n t\right)e^{-\omega_n t}$$

$$\dot{x} = \omega_n \left(-x_0 + (1 - \omega_n t)\left(\frac{v_0}{\omega_n} + x_0\right)\right)e^{-\omega_n t}$$

$$0 = -x_0 \; (1 - \omega_n t) \left(\frac{v_0}{\omega_n} + x_0\right)$$

$$= -\omega_n t \; x_0 - v_0 t + \frac{v_0}{\omega_n}$$

$$t = \frac{v_0}{\omega_n \; (\omega_n x_0 + v_0)}$$

$$x_{max} = \left(x_0 + \frac{v_0}{\omega_n}\right)e^{-\left(\frac{v_0}{(\omega_n x_0 + v_0)}\right)}$$

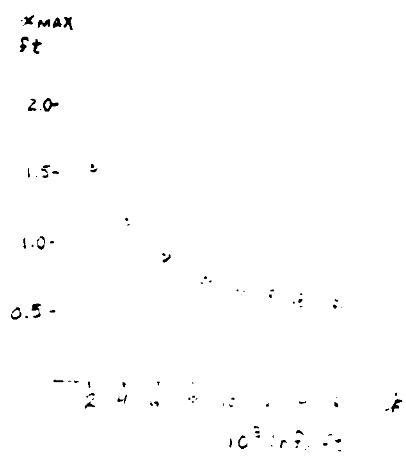
with zero initial displacement

$$x_{max} = \frac{v_0}{\omega_n} e^{-1}$$

$$= .366 \; \frac{v_0 \; (m)^{1/2}}{(k)^{1/2}}$$

$$= (.366) \; (16) \; \left(\frac{4580}{32.174}\right)^{-1/2} \; (k)^{-1/2}$$

$$x_{max} = 69.8 \; (k)^{-1/2}$$



Spring-Displacement Chart

for a one foot stopping distance

$k = 5000$ lbf/ft

$c = 2\ (km)^{1/2}$

$= 2\ ((5000)\ \frac{(4580)}{(32.174)})^{1/2}$

$c = 1687$ lbf-s/ft

$$\dot{x} = \omega_n (-x_o + (1 - \omega_n t)\ (\frac{v_o}{\omega_n} + x_o))e^{-\omega_n t}$$

$$\ddot{x} = \omega_n\ (-\omega_n(\frac{v_o}{\omega_n} + x_o)\ \ e^{-\omega_n t} - \omega_n(-x_o + (1 - \omega_n t)\ (\frac{v_o}{\omega_n} + x_o))e^{-\omega_n t}$$

$$= \ (-\omega_n^{\ 2}\ (\frac{v_o}{\omega_n} + x_o) - \omega_n^{\ 2}\ (-x_o + (\frac{v_o}{\omega_n} + x_o - v_o t - x_o\omega_n t))e^{-\omega_n t}$$

$$= -\omega_n^{\ 2}\ (2\frac{v_o}{\omega_n} + x_o - v_o t - x_o\omega_n t)e^{-\omega_n t}$$

$$\ddot{x} = -\omega_n^2\ ((\frac{2}{\omega_n} - t)\ v_o + (1 - \omega_n t)\ x_o)e^{-\omega_n t}$$

with $x_o = 0$

$$\ddot{x} = (\omega_n^{\ 2} t - 2\omega_n)v_o e^{-\omega_n t}$$

the force in the alignment arm is then

$$f = m\ddot{x}$$

$$f = m(\omega_n^{\ 2} t - 2\omega_n)\ v_o e^{-\omega_n t}$$

$$f = (\frac{4580}{32.174})\ (\frac{(5000)\ (32.174)}{4580}\ t - 2(\frac{(5000)\ (32.174)}{4580})^{1/2})\ (16)e^{-(\frac{(5000)(32.174)}{4580})^{1/2} t}$$

$$f = (2275\ (35.1t - 2(5.92))e^{-5.92t}$$

$$x = v_o t e^{-\omega_n t}$$

$$x = 16te^{-5.92t}$$

$$f = (79{,}900t - 26{,}900)e^{5.92t}$$

APPENDIX B

SPREADER BAR

ALTERNATIVE CONCEPTS FOR SOLUTION OF DYAMIC ALIGNMENT PROBLEMS

SPREADER BAR

ALTERNATIVE CONCEPTS FOR SOLUTION OF DYNAMIC ALIGNMENT PROBLEM

TABLE OF CONTENTS

1.0 INTRODUCTION

1.1 General

Spreader bars (see figure 1) are structurally strong metal devices, of three different dimensions to be compatible with 20-, 35-, and 40-foot containers. The spreader bar, suspended from a crane at the four lift lug attachments, is lowered onto a container top, where four bayonet studs on the spreader bar are inserted simultaneously into the four bayonet-accepting holes in the container top.

Spreader bars are utilized to facilitate transfer of military ISO containers during freight loading and unloading operations. When it is necessary to perform these operations at sea, dynamic alignment problems can occur even during relatively mild sea state conditions. Although these problems are an ongoing concern for standard commercial operations, the need to solve the dynamic alignment problem is urgent during military operations where rapid transfer is a necessity.

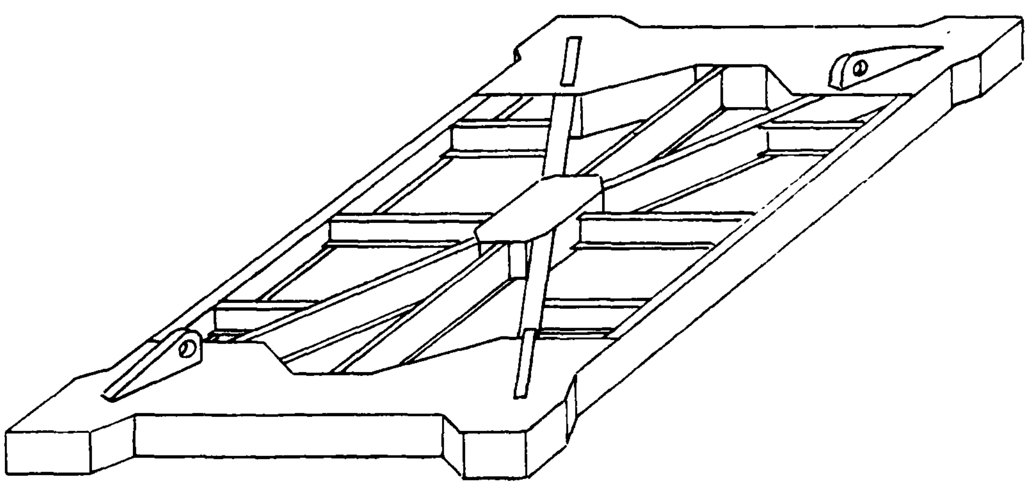


Figure 1. Spreader Bar, Lifting ISO and Freight Containers

1.2 Approach

The solution to the dynamic alignment problem can be approached systematically from either a passive or active viewpoint. The active system approach would require that the crane operator be considered part of the system, i.e., the operator would receive feedback with respect to the dynamic state and relative position of the spreader bar and container and would make appropriate changes to the dynamic state and relative position through system controls. The passive system approach would minimize or eliminate the active participation of the crane operator beyond lowering the spreader bar and positioning it such that its motion is centered over the container

1.2.1 Active System

An example of an active system would be one where a directional thrusting device is attached to the spreader bar and operated from the crane cabin using a joy stick or similar type of direction and thrust control. Visual feedback would be utilized in positioning the control in order to minimize the motion of the spreader bar.

Because the active system is inherently susceptible to human error, only the passive system will be considered in this document.

1.2.2 Passive Systems

The passive systems considered here are of two types. Both types utilize a broad definition of alignment arms. One type of passive system utilizes rigid alignment arms which are attached to the spreader bar through an energy absorbing mechanism. The other type utilizes flexible alignment arms (stranded wire rope cables) directly attached to the container and indirectly attached to the spreader bar such that the spreader bar is drawn down the cables to the container.

Three passive system concepts are analyzed in this report. The concepts, utilizing rigid alignment arms, are described in section 2.0. An evaluation of the concepts is provided in section 3.0. Section 4.0 provides a summary of the conclusions drawn from the analyses and a recommendation for implementation of one of the concepts.

2.0 DESCRIPTION OF CONCEPTS

2.1 Scissors Action Alignment Arms

The scissors action alignment arm is depicted on the 20-foot spreader bar in figure 2.The device is spring loaded at the center. The sliding feet of the scissors provide coulomb damping. The alignment arms are hinged at three points; in the up position they do not extend beyond the overall plan outline of the spreader bar. In the down position, the alignment arm extends 17 inches below the elevation outline of the spreader bar. During complete extension, the total travel of the alignment arm on the 20-foot side is 3 feet. The alignment arm on the eight-foot side has a 1.5-foot total travel. The widths of the respective scissors action alignment arms are 10 and 5 feet. It is a welded assembly of primarily 4" x 4" x ¼" angle steel.

Figure 3 depicts (a) the braking mechanisms attached to the scissor legs, (b) the hold-down flange for the scissors pin, and (c) the stowage mechanism. The hold-down flange engages the scissors pin between the positions of its extreme travel. At either extreme, the alignment arm is free to rotate up about the hinges. The shape of the alignment arms is such that backside or side impacts will cause it to rotate up about the hinges.

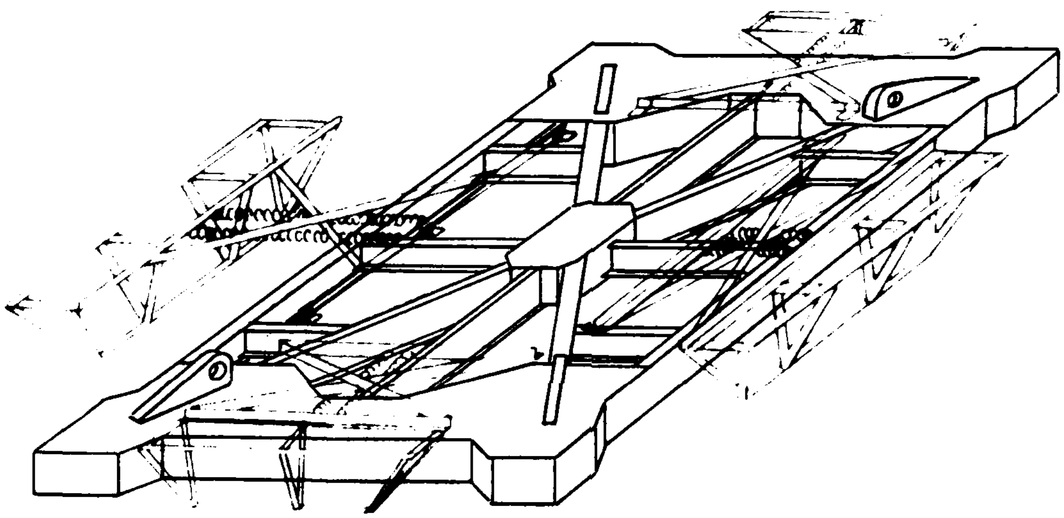


Figure 2. Scissors Action Alignment Arms

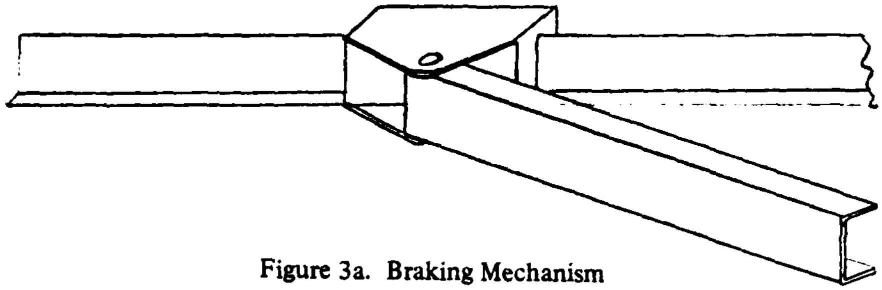


Figure 3a. Braking Mechanism

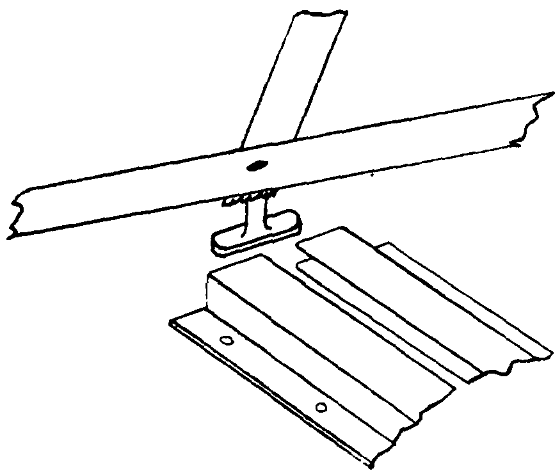


Figure 3b. Scissors Pin Hold-Down Flange

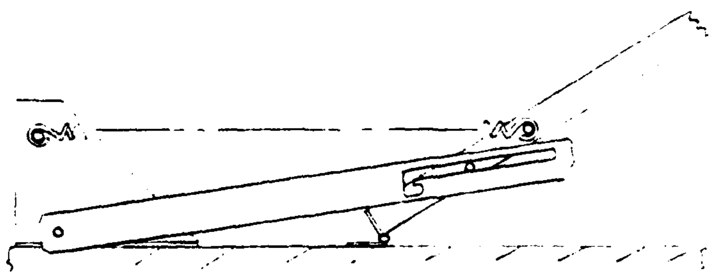


Figure 3c. Stowage Mechanism

Operationally, one of the alignment arms on the short side and one on the 20-foot side are secured in the up position. As the spreader bar passes over the container, either one or both of the other alignment arms engage the container slowing the spreader bar first then reversing its motion as the alignment arms return to their original position effectively aligning the spreader bar for engagement with the container.

2.2 Spring/Damper Alignment Arm No. 1

The spring/damper alignment arms are energy absorbing devices that utilize a commercial combination spring/damper assembly. Figure 4 depicts the devices mounted on a 20-foot spreader bar. Two alignment arms are mounted on each long side of the spreader bar, while one is mounted on each short side. The alignment arms are 3-inch diameter heavy wall steel tubes. They do not extend beyond the spreader bar's plan outline in either the stowed position or the ready position, as depicted.

The tube is attached to the spring/damper through a double clevis welded to the tube. A spring loaded tang, as indicated in figure 5, engages the spring/damper shaft eye to inhibit relative angular movement until the disengagement occurs at a predetermined torque. Disengagement also occurs when a predetermined vertical force impinges on the bottom of the alignment arm.

The alignment arm is a two-part assembly. The two components are hinged together at the bottom of the spreader bar with a torsion spring return hinge to allow it to rotate under the bar if it is impacted from its backside. If the arm is impacted from the side, the spring/damper shaft will rotate allowing the alignment arm to swing free from the impact source.

The alignment arm is placed in its stowed position by rotating it about the spring/damper shaft past a wedge-shaped block welded to the side of the spreader bar. The spring in the spring/damper acts to hold the device in the stowed position. The tool depicted in figure 6 is used to place the alignment arm in either the stowed or ready position.

The spring/damper is secured to a plate welded to the spreader bar by a "U" bolt capturing the shaft and by a bolt through the rear eye.

2.3 Spring/Damper Alignment Arm No. 2

The second spring/damper alignment arm is functionally similar to the first device described in paragraph 2.2. Figure 4 is representative of the six devices mounted on a 20-foot spreader bar. The spring/damper assembly can be heavier than that used with spring/damper alignment arm No. 1 because the moment transmitting mechanism is heavier structurally. The difference between the two alignment arms is the method of providing release mechanisms for avoiding destructive forces on the alignment arms, spreader bar, or container.

The first alignment arm utilizes springs which can be set to release at predetermined force or torque levels. The second alignment arm utilizes displacement to affect the necessary releases.

As indicated in figure 7, the alignment arm's upper tube has short rods welded to it that engage a hold-down flange when the alignment arm is between its extreme positions. In the ready position, the alignment arm can rotate about the spring/damper shaft when impacted from the side and rotate about the clevis pin when impacted vertically from below. At the maximum travel of the spring/damper shaft, the rods on the alignment arm disengage from the hold-down flange allowing rotation about the clevis pin. This disengagement precludes component damage caused by overloading the spring/damper assembly.

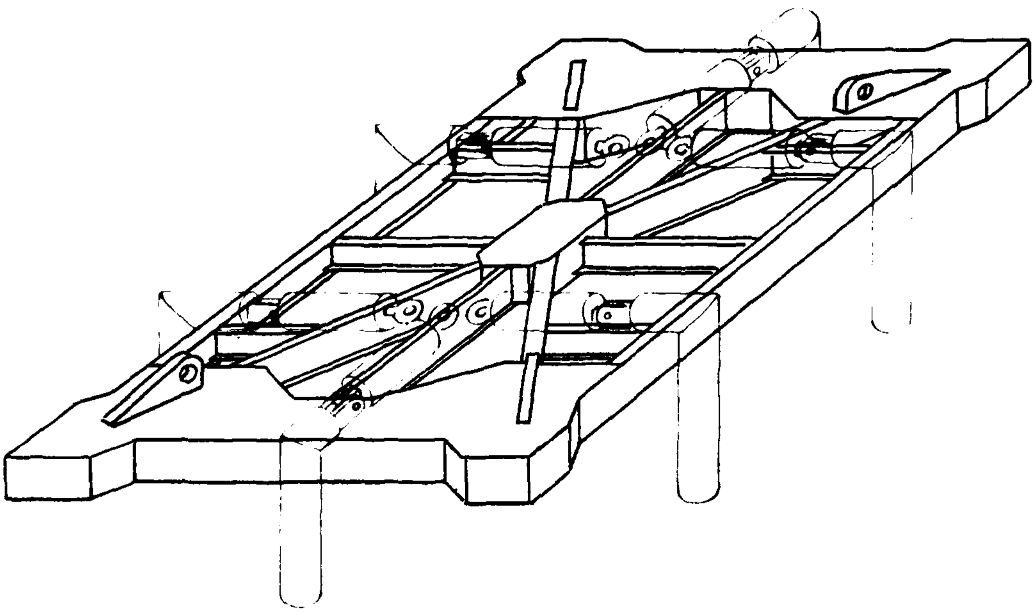


Figure 4. Spring/Damper Alignment Arms

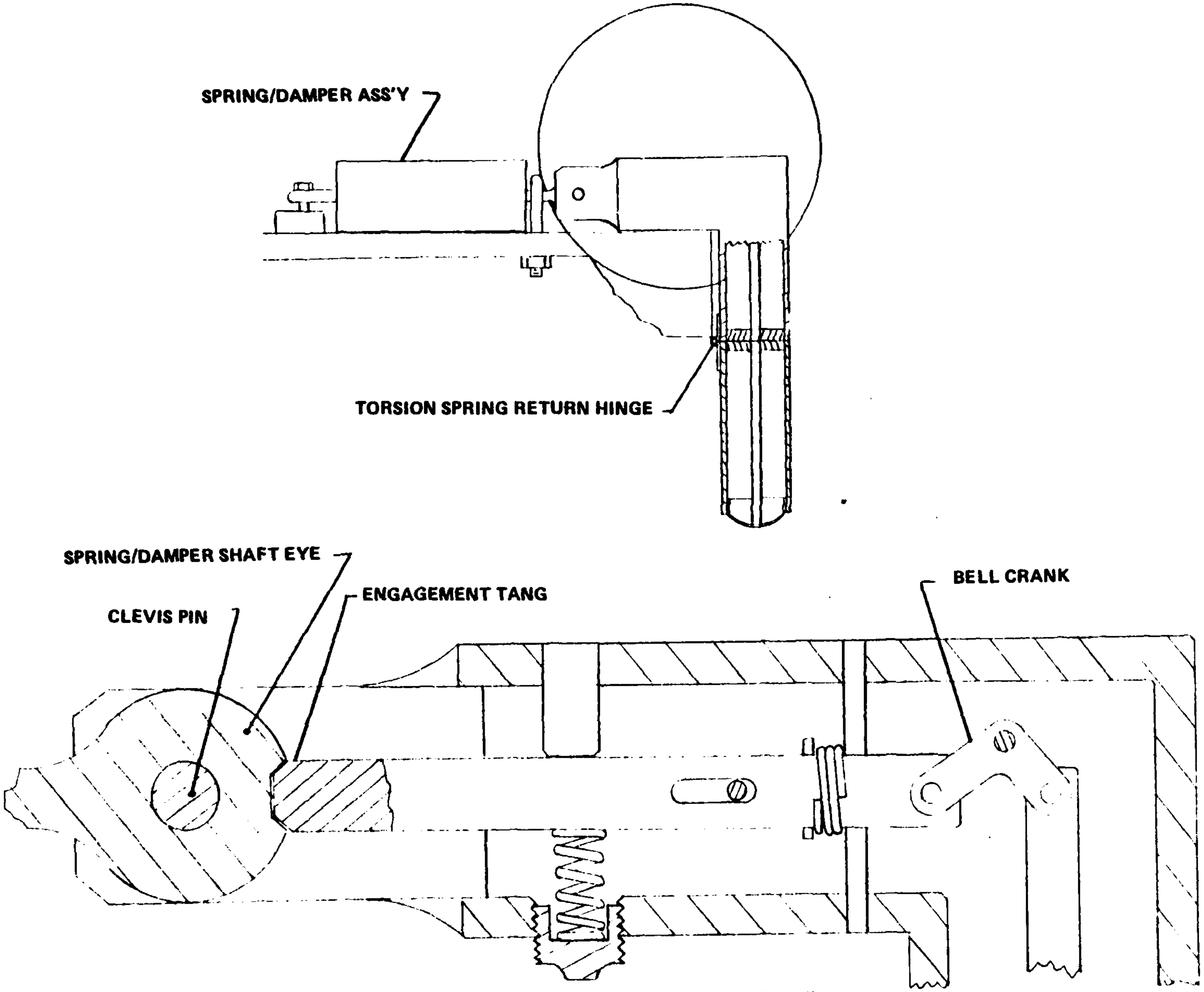


Figure 5. Spring/Damper Alignment Arm #1 Detail

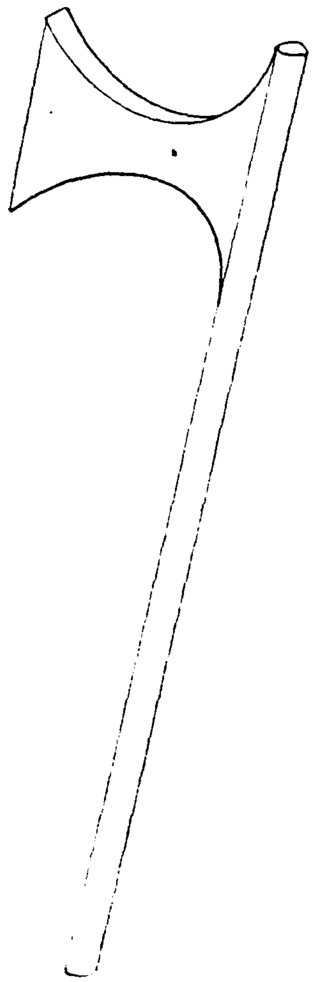


Figure 6. Spring/Damper Alignment Arm Stowage Tool

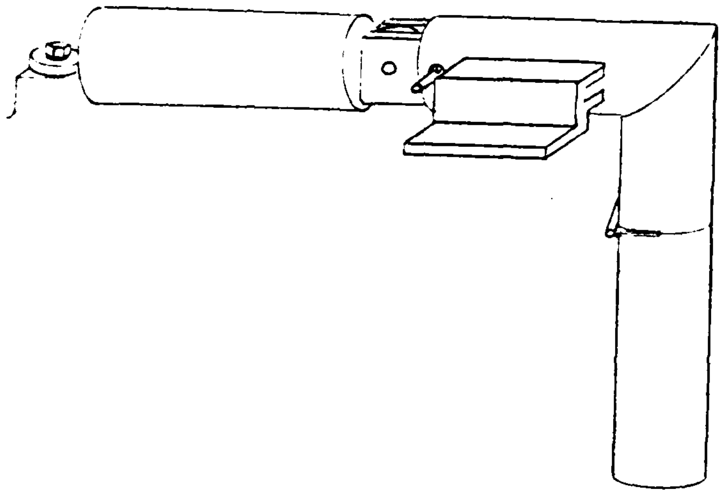


Figure 7. Spring/Damper Alignment Arm #2 Detail

Backside impacts cause release of the hinged portion of the alignment arm below the spreader bar, as was the case with spring/damper alignment arm No. 1.

The second alignment arm is placed in the stowed position the same way the first spring/damper alignment arm is placed in the stowed position.

3.0 EVALUATION OF CONCEPTS

3.1 Capability to Perform Primary Mission

1. The scissors action alignment arm has the greatest capability for absorbing energy since its displacement potential is three feet.

2. The spring/damper alignment arm No. 1 can be displaced one foot, therefore, compared to the scissors action arm it requires three passes to absorb the same energy.

3. The spring/damper alignment arm No. 2 has the same displacement capability of spring/damper No. 1 but can absorb more energy because it can use a heavier spring/damper.

3.2 Capability for Positioning Between Stowed and Ready Position

1. The scissors action alignment arm is spring loaded to assist in positioning. The mechanism for holding and releasing the stowed position is relatively more complex than the other two concepts and under impact and vibration it is less reliable.

2. Spring/damper arm No. 1 is stowed very simply as previously indicated.

3. Spring/damper arm No. 2 is identical to No. 1 in its stowage capability.

3.3 Capability of Withstanding Backside Impacts

1. The scissors action alignment arm, being spring loaded, rises up with backside impacts. Therefore its capability is relatively good.

2. Spring/damper No. 1 is hinged to swing under the spreader bar; thus it can be caught between the container and the spreader bar. This would require the crane operator to raise the spreader bar a little.

3. Spring/damper No. 2 is identical to No. 1 in withstanding backside impacts.

3.4 Transmission of Vertical Forces to the Container Top

1. The scissors action device will rise up when impacted from below with vertical forces. It will transmit very low forces to the container top.

2. Spring/damper No. 1 will also rise up when impacted from below and will also only transmit low forces to the container top.

3. Spring/damper No. 2 rises up when impacted from below and also will transmit only low forces to the container top.

3.5 Capability of Withstanding Side Impacts

1. The scissors action alignment arm rises up when impacted from the side.

2. Spring/damper No. 1 will rotate up under side impacts.

3. Spring/damper No. 2 will rotate up under side impacts.

3.6 Relative Operational Complexity

1. The scissors action alignment arm is operationally simple. The only difficulty is in positioning it in the ready position from the stowed position. and that backside or bottom impacts can place it in the stowed position.

2. Spring/damper No. 1 is also operationally simple. It can be put in the stowed position with side impact. but it can be easily replaced in the ready position.

3. The operational complexity of spring/damper No. 2 is the same as that of No. 1.

3.7 Relative Mechanical Complexity

1. The scissors action alignment arm's complexity is minimal as long as its structural integrity is maintained.

2. The spring/damper No. 1 is fairly complex mechanically, but the protection afforded to the mechanisms minimizes the effect of the complexity.

3. The spring/damper No. 2 is very simple mechanically.

3.8 Exposure of Functional Mechanisms to Elements

1. The scissors action mechanisms are exposed to environmental elements.

2. The spring/damper No. 1 has its mechanisms completely protected.

3. The spring/damper No. 2 has its mechanisms exposed to the environment but its simplicity offers no opportunity for environmental damage.

3.9 Structural Integrity

1. The scissors action mechanism is most suspect structurally since improper loading could cause bending of the scissors arms causing damage that would require emergency maintenance.

2. The spring/damper No. 1 is structurally sound since its mechanisms are protected by being internal. The hinge requires special structural consideration since it is heavily loaded during normal operations.

3. The spring/damper No. 2 is sound structurally, and can be more heavily loaded.

3.10 Estimated Fabrication Costs

1. Scissors action:

   Relatively expensive because of the quantity of structural elements and welding operations.

2. Spring/damper No. 1:

   Less expensive than the scissors action device because of redundant components and less material, but the internal components increase its cost above spring/damper No. 2.

3. Spring/damper No. 2:

   Inexpensive redundant components.

4.0 CONCLUSIONS AND RECOMMENDATIONS

4.1 Conclusions

Table 1 evaluates the various systems as good, fair, or poor in accomplishing tasks or meeting essential criteria. The scissors action mechanism's exposure to the elements and the subsequent susceptibility to damage are overriding considerations. Spring/damper arm No. 1 cannot absorb as much energy as No. 2. It is complex mechanically and more expensive as well. This analysis indicates that system three, spring/damper No. 2, is the most viable concept.

4.2 Recommendation

It is recommended that system three, spring/damper No. 2, be accepted for fabrication and testing.

Table 1. Alignment Arm Concept Evaluation

	Scissors Action	Spring/Damper No. 1	Spring/Damper No. 2
Capability to perform primary mission	Good	Poor	Fair
Capability for positioning between stowed and ready	Fair	Good	Good
Capability of withstanding backside impacts	Good	Fair	Fair
Transmission of vertical forces to the container top	Good	Good	Good
Capability of withstanding side impacts	Good	Good	Good
Relative operational complexity	Fair	Good	Good
Relative mechanical complexity	Fair	Poor	Good
Exposure of functional mechanisms to elements	Poor	Good	Good
Structural integrity	Poor	Fair	Good
Estimated fabrication costs	Poor	Fair	Good

APPENDIX

DISCUSSION OF SEVERAL ALTERNATIVE ALIGNMENT ARM CONCEPTS

1. Figure A1 shows a concept currently being evaluated. A radio-controlled spreader bar with diesel driven hydraulically actuated alignment arm is used. Improvements can be made to the alignment arm design with respect to side, backside, and vertical loading.

2. Figure A2 shows a flexible alignment arm concept in which four motors drive winches whose cables align and take up slack as the crane operator lowers the spreader bar. The ends of the cables are attached to a lightweight frame which is secured to the ISO container.

3. Figure A3 shows another flexible alignment arm concept in which a motor drives four pulleys which roll the spreader bar along four cables. One end of each cable is attached to the crane while the other end of each cable is fixed to a lightweight frame which is secured, in turn, to the ISO container.

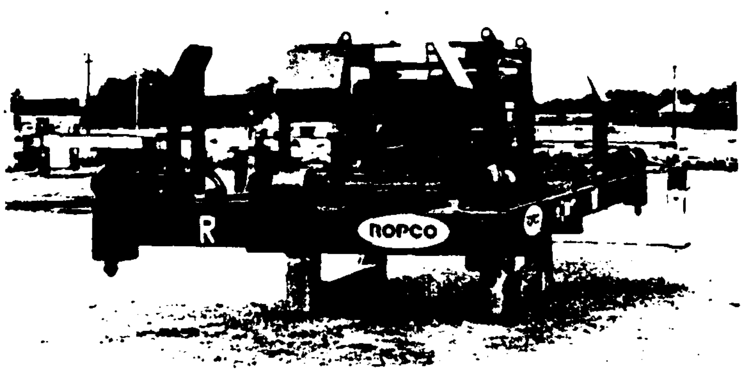


Figure A1. Radio-Controlled Spreader Bar

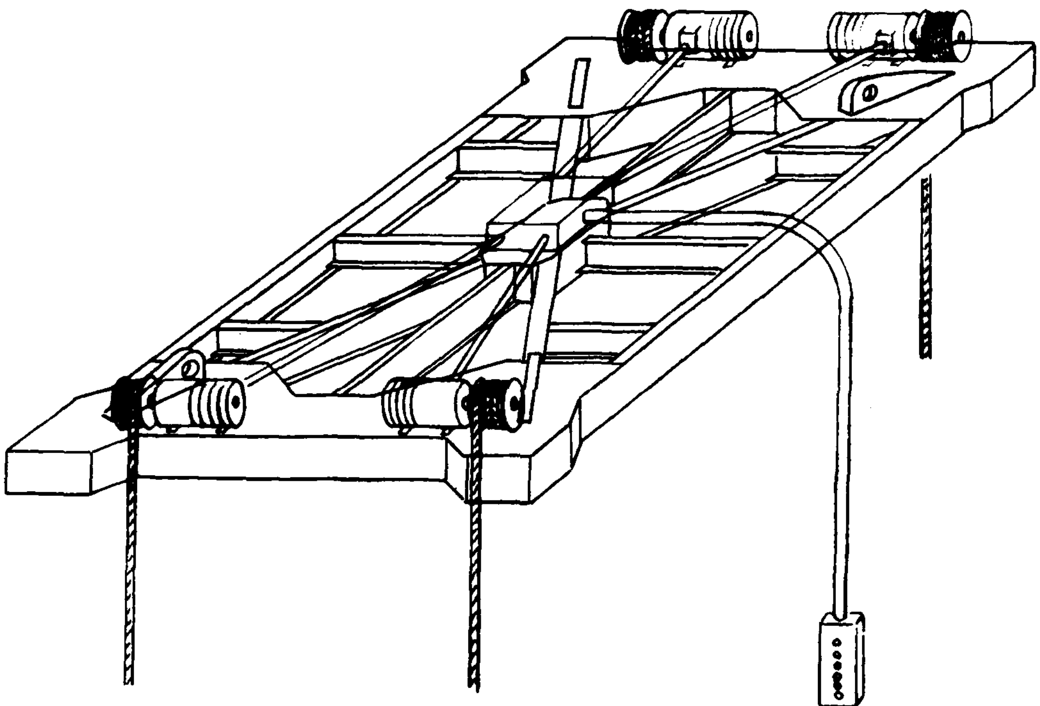


Figure A2. Winch Drive Spreader Bar

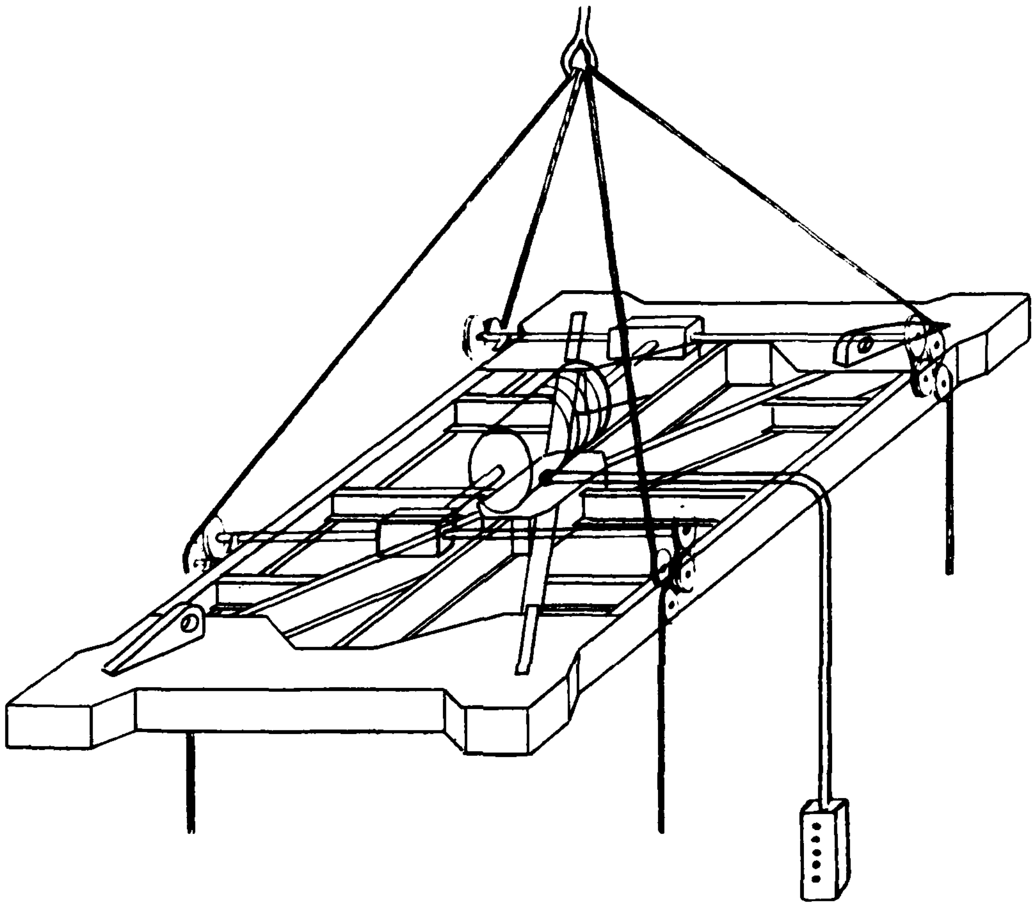


Figure A3. Roller Drive Spreader Bar

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