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Brownell et al.

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[54] INDICATOR OR CONTROL FOR CRANES

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Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 799,739, Feb. 17, 1969, abandoned.

[52] **U.S. Cl.**.....**212/39 MS**, 212/39 R, 340/267 C

[51] **Int. Cl.**.....**B66c 13/48**

[58] **Field of Search** .....212/39, 39 MS; 340/267 C

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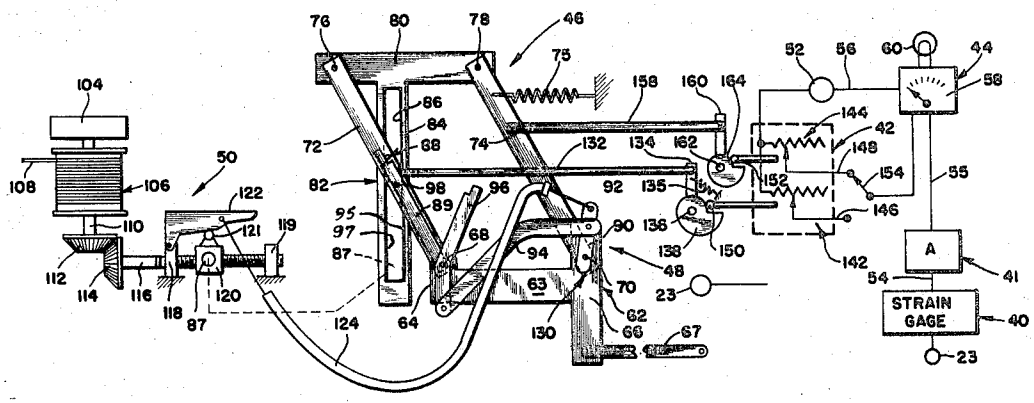
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[57] ABSTRACT

A safe load indicator for a mobile crane including an extensible boom, the indicator including a strain gage means to provide an electrical signal reflecting total load moment about the boom horizontal pivot axis and linkage means moveable in accordance with boom movement to modify the signal according to boom vertical angle and length. To achieve an accurate indication of maximum permissible load moment throughout the full operational range of the boom, linkage movement modifying means is provided to interrupt or reverse linkage movement responsive to departure of the boom from an operational range wherein crane stability is critical and entry into an operational range wherein structural strength of the crane components is critical.

4 Claims, 6 Drawing Figures



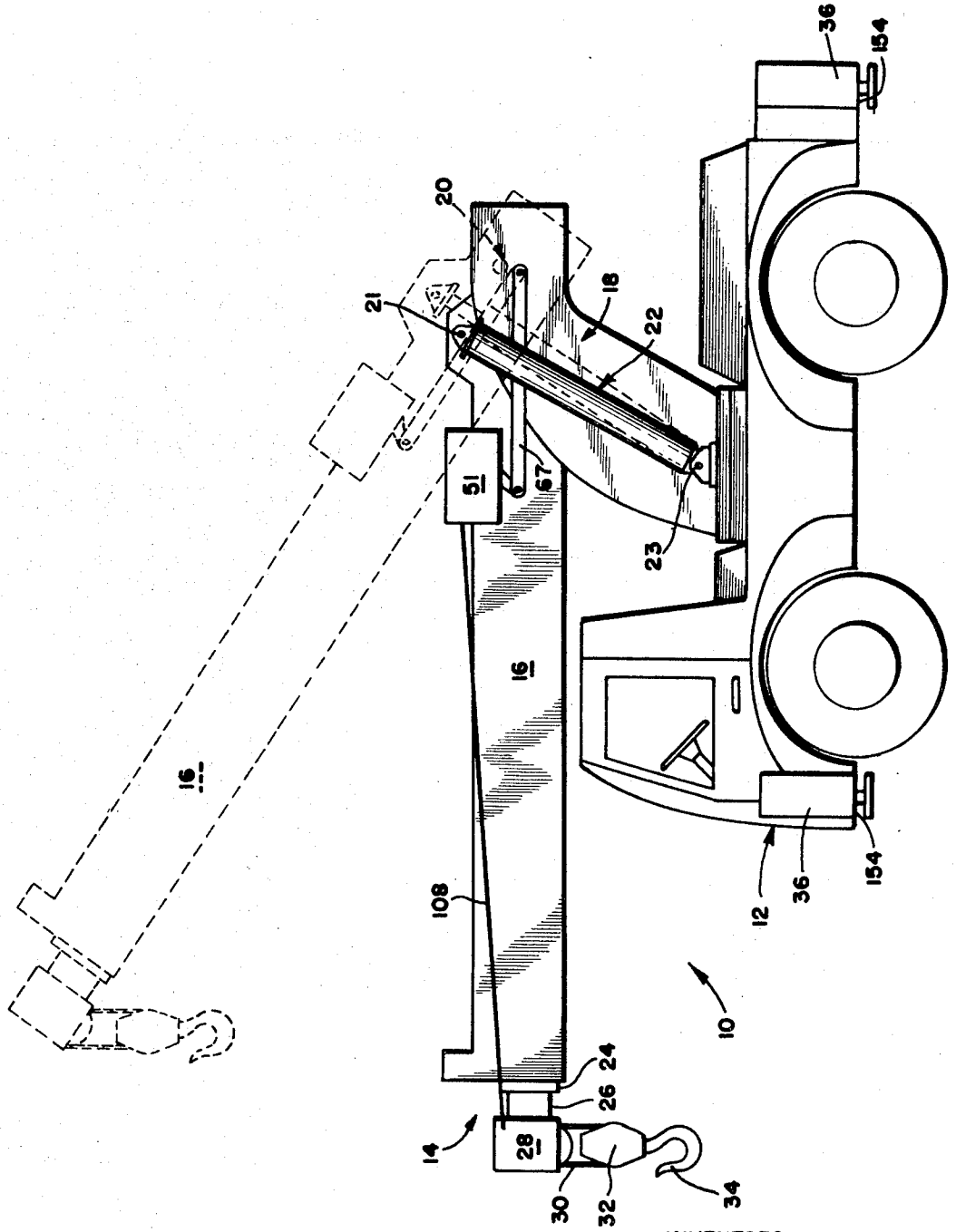


FIG. 1

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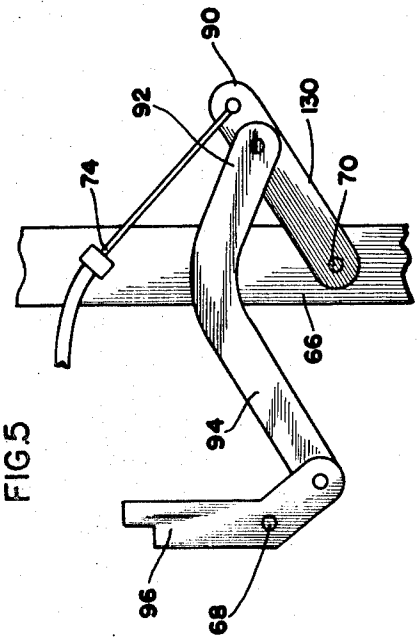


FIG.5

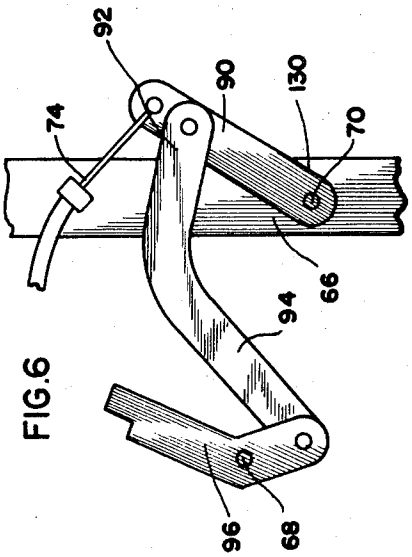


FIG.6

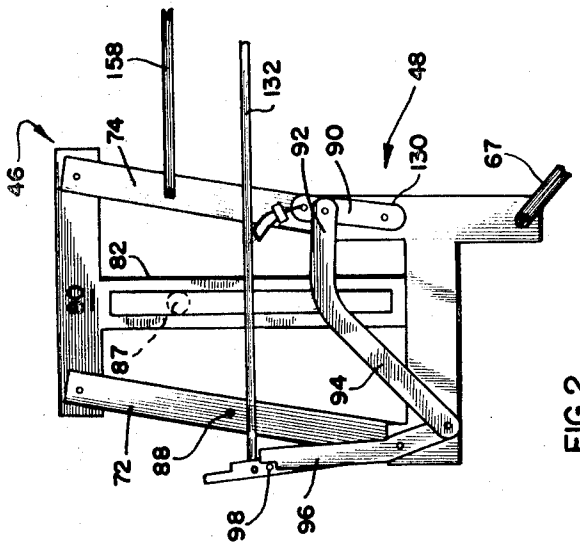


FIG.4

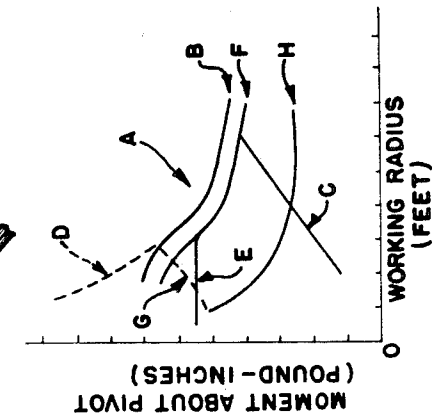
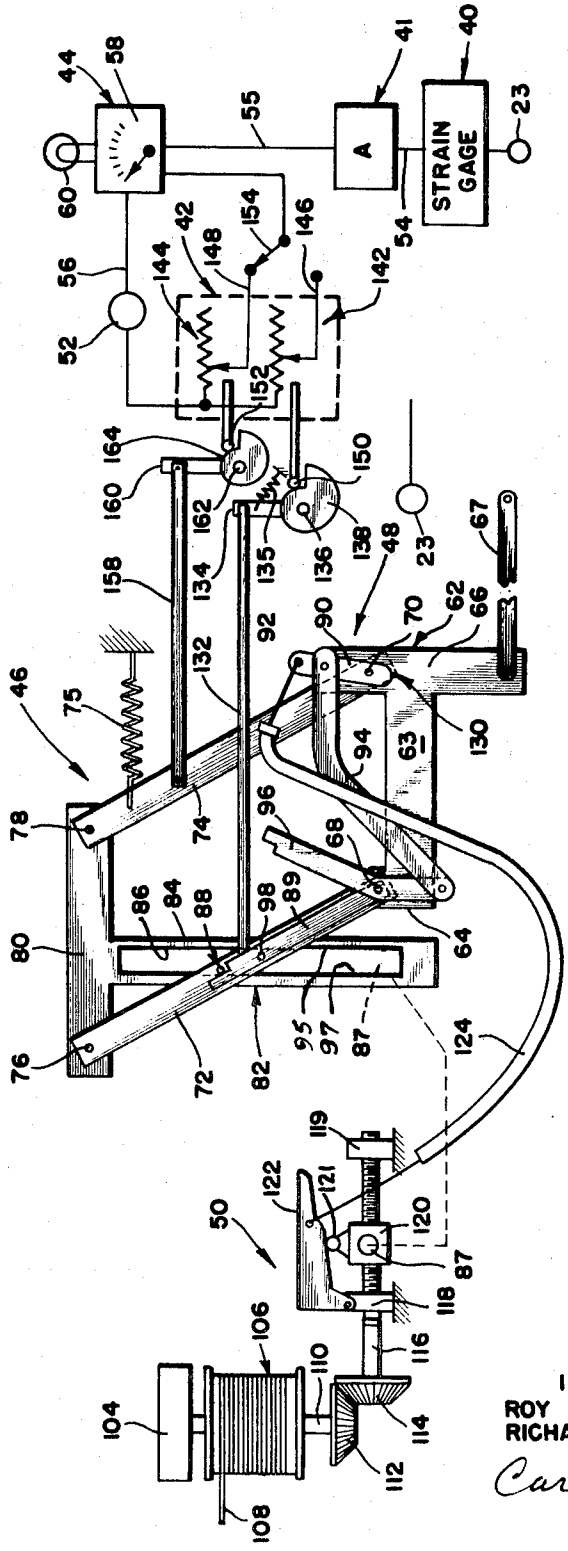


FIG.2

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FIG.3



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INDICATOR OR CONTROL FOR CRANES

BACKGROUND OF THE INVENTION

This application is a continuation in part of co-pending application Ser. No. 799,739 entitled "INDICATOR OR CONTROL FOR CRANES" filed Feb. 17, 1969 now abandoned. This invention relates to cranes and more particularly to indicators or controls for cranes.

Heretofore, safe load indicator and control devices have been available for vehicle-mounted cranes to signal an overload condition which might cause tipping of the crane or failure of the crane components. Such devices may include a load gage to measure line load, a signal from which may be modified according to boom angle and length to provide a read-out of load moment or a warning signal. Such devices have not been generally satisfactory because they fail to simply and accurately establish and compensate for all the parameters determinative of total or effective load moment. These parameters include line or hook load, boom vertical angle and length, vehicle and load orientation, and parasitic loads such as those variously induced by boom deflection, line reaving and environmental conditions. Consequently, to compensate for unconsidered or inaccurately established parameters, it has been the practice in establishing maximum or safe permissible loads to include a substantial safety factor thereby prohibiting consistent utilization of the full capacity of the crane in many operational conditions. Other prior art devices have endeavored to enhance accuracy by attempting to provide a read-out of actual or total load moment exerted by the boom on the vehicle and to modify the read-out according to boom vertical angle. Such devices may include a fluid pressure gage means to measure fluid pressure at the pressure side of the boom hoist or lift cylinder. However, these latter devices have not, in practice, uniformly achieved accuracy because of errors induced by uncontrollable variations in fluid pressure at the low pressure side of the boom lift cylinder. Additionally, these devices have failed to accurately indicate maximum permissible load moment through all the operational ranges of the boom because they have failed to consider the effect of parasitic loads which vary as to boom length. Still further, such devices have failed to operationally distinguish between the operational range of the boom wherein maximum load moment is limited by loads which would cause vehicle tipping, as opposed to other operational ranges wherein crane stability is not critical but wherein other limitations such as structural strength of the crane components are critical. For example, if a loaded crane boom is in a relatively low position, crane tipping and not structural strength may be the critical consideration. However, if the boom is pivoted to a relatively high position, the effective boom working radius or load lever or moment arm is reduced. Consequently, a load which would not be critical from a stability standpoint, may become critical from a structural standpoint by virtue of the compressive loads generated which may cause failure of the boom components.

BRIEF DESCRIPTION OF THE INVENTION

It is a general object of the present invention to provide an indicator or control device for cranes which

eliminates or circumvents the problems heretofore discussed.

In achieving this general object, the present invention provides a boom orientation responsive device for a material handling apparatus including a boom mounted on a vehicle for pivotal movement about a horizontal axis. The vehicle is adapted to be supported on a supporting surface and the boom is adapted to support a load on a free end thereof at a plurality of positions having a plurality of moment arms about the horizontal axis. The device includes boom orientation means for providing a signal responsive to a change in the moment arm about the horizontal axis and moveable means operatively connected to the boom orientation means to actuate the same. The moveable means is moveable in a first direction in accordance with the change in the length of the moment arm about the horizontal axis during movement of the boom in a first direction. To reflect a change in the maximum permissible load moment induced by movement through an operational range wherein one load limiting consideration is critical, into another operational range wherein another load limiting consideration is critical, movement modifying means are provided for modifying movement of the said moveable means during further movement of the boom in the first direction to alter actuation characteristics of the boom orientation responsive means.

In one form of the present invention, strain gage means are provided for reading total moment exerted by the boom about the horizontal pivot axis and the device may further include the boom extension means responsive to longitudinal extension and retraction of the boom for moving said moveable means.

BRIEF DESCRIPTION OF THE DRAWINGS

A better understanding of the present invention may be had by reference to the accompanying drawings wherein:

FIG. 1 is a schematic elevational view of a mobile crane adapted to be provided with a safe load indicator or control according to the present invention;

FIG. 2 is a typical crane load moment chart or graph reflecting maximum permissible load moment on a crane through various operational ranges and conditions, boom effective working radius in feet being plotted against moment about the boom pivot axis in inch pounds;

FIG. 3 is a schematic, partially exploded illustration of a safe load indicator according to the present invention, adapted to be utilized with the mobile crane shown in FIG. 1;

FIG. 4 is a schematic illustration of the parallel linkage assembly of the linkage position corresponding to an elevated position of the boom;

FIG. 5 is a schematic illustration of the linkage movement modifying mechanism shown in FIG. 4 but showing the mechanism position corresponding to a retracted length of the telescopic boom; and

FIG. 6 is a schematic illustration of the linkage movement modifying mechanism shown in FIG. 5 but showing the mechanism position corresponding to an extended length of the telescopic boom.

DETAILED DESCRIPTION OF THE PRESENT INVENTION

Referring in more detail of FIG. 1 of the drawings, a safe load indicator according to the present invention, is adapted for use on a conventional mobile, self-propelled crane **10**. The crane includes a motorized wheeled vehicle **12** on which is supported a longitudinally extensible boom **14** mounted for pivotal movement about vertical and horizontal axes. The boom assembly **14** comprises a boom base section **16** pivotally connected to a boom support or shipper **18** for pivotal movement about a horizontally disposed shipper pivot pin **20**. An extensible fluid motor or hydraulic cylinder **22** is pivotally connected between the boom base section **16** and the boom support **18** by upper and lower cylinder pivot pins **21** and **23** for pivotally moving or elevating the boom assembly **14** in a vertical plane. The boom support **18** is pivotally connected to the vehicle **12** for pivotable movement about a vertical axis which may be effected by a rotary fluid motor, not shown. The boom assembly **14** further includes first and second extensible sections **24** and **26**, telescopically received in the base section **16** and adapted to be longitudinally extended or retracted by extensible fluid motors, not shown. It should be noted that while two extensible sections are shown, any number of extensible sections may be provided. A conventional manually operable fluid control, not shown, may be provided for controlling boom elevation and extension. On the outer end of the second extensible section **26**, a boom point or fixed sheave block **28** is provided over which a cable **30** is trained for suspending a moveable sheave block **32** supporting a load engaging hook **34**. Conventional laterally and downwardly extensible outriggers, **36** may be adjustably mounted on the sides of the vehicle **12** for selective contact with the ground for an added stabilizing effect during certain lifting operations.

In operating mobile cranes of this type, the consideration limiting maximum line load when the boom assembly **14** is in a relatively low lifting position or operating range, adjacent that shown in full lines in FIG. **1**, is usually crane stability against tipping. A typical crane maximum permissible load moment chart is shown in FIG. **2** wherein boom effective working radius in feet is plotted against moment about the boom pivot axis in pound inches. As the boom assembly **14** is elevated or retracted, the effective working radius or load lever or moment arm decreases and maximum permissible load moment increases without inducing tipping, as illustrated by segment A of curve B of the load chart shown in FIG. **2**. Parasitic load moment decreases as working radius decreases, as indicated by curve C. If stability were the only limiting factor, maximum permissible load moment would continue to increase as working radius decreases, as illustrated in segment D of curve B. However, when the boom assembly **14** has been positioned to attain a relatively short working radius, as elevated passed that shown in phantom lines in FIG. **1**, the factor limiting maximum permissible load moment is structural strength of the crane components. It is of importance that structural strength of the components may be substantially less than the maximum permissible load from a stability standpoint. Further, in some instances maximum structural load moment in this shortened working radius range may remain constant as working radius decreases, as indicated in segment E of curve F. In other instances maximum structural load moment may decrease as the working radius decreases, as indicated in segment G of curve B. It has been found that the reversal point wherein maximum permissible load moment ceases to increase as working radius decreases and commences to decrease, varies as to boom length. For this reason, a number of such reversal points exist for an extensible or telescopic boom. Additionally, it has been determined that in some cases the rate at which maximum permissible load moment decreases as working radius decreases in the working range wherein structural strength is critical (segment G of curve B) approximates a reverse of the rate that maximum permissible load moment increases as working radius decreases in the working range wherein stability may be critical (upper portion of segment A). Based on these concepts, the present invention provides a safe load indicator which not only compensates for all of the parameters determinative of actual crane loading but also provides a means to accurately establish maximum permissible load moment in both the working range wherein stability is critical and as well as the working range wherein structural strength is critical.

With reference to FIG. **3** of drawings, the present invention provides a unique safe load indicator, the basic components of which may be summarized as follows. To read total or effective load moment exerted on the vehicle **12** by a load, as well as that induced by parasitic loads, a strain gage assembly **40** is operatively connected to the lower boom lift cylinder pivot pin **23** to read the strain induced therein. The signal output of the strain gage assembly is directed to an amplifier assembly **41** and to an indicator assembly **44**. A potentiometer assembly **42** is connected to a boom orientation responsive linkage assembly **46** which adjusts the potentiometer assembly according to working radius as determined by boom angle and length. A signal from the potentiometer assembly **42** is accordingly directed to the indicator assembly **44** which compares this latter signal with that from the strain gage assembly **40** and indicates a percentage of maximum permissible load moment. The linkage assembly **46** includes a linkage movement modifying mechanism **48** which is operatively connected to a boom extension responsive assembly **50** to adjust the point at which linkage movement is modified according to boom length. These assemblies may be positioned in a housing **51** fixed on the boom base section **16**.

More specifically, the strain gage assembly **40**, which may be of a type similar to the SR-4 strain gages produced by BLH Electronics, is intended to provide an electrical signal the magnitude of which is proportional to total load moment on the boom assembly **14**. The strain gage assembly is connected to the amplifier assembly **41**, the indicator assembly **44** and a suitable source of electric current, such as the vehicle ignition switch **52** by electric leads **54**, **55** and **56**. The indicator assembly **44** may include an electric calibrated meter **58** for indicating a percentage of permissible load moment about the pivot axis, and an electric light **60** to warn of an overload condition.

The boom orientation responsive linkage assembly 46 comprises a support arm 62 pivotally supported on the boom base section 16 by a pivot pin 70. The support arm 62 includes a horizontal leg 63 fixed to first and second vertical legs 64 and 66, respectively. The lower end of the second leg 66 is pivotally connected by a shipper link 67 to the boom support or shipper 18 below the pivot pin 20. The upper ends of the first and second vertical legs 64 and 66 are pivotally connected by pivot pins 68 and 70 to the lower ends of generally vertical first and second parallel linkage arms 72 and 74. The upper ends of the parallel linkage arms 72 and 74 are spring-loaded by a spring 75 to pivot in a clockwise direction, as shown in FIG. 3, and are connected by pivot pins 76 and 78 to opposite ends of a horizontal leg 80 of a T-shaped cam follower bar assembly 82. The cam follower bar assembly 82 further includes a vertical leg 84 having vertically extending surfaces 95 and 97 forming a slot 86 therein. The slot 86 freely receives a pin 87 forming a part of the boom extension responsive assembly 50. Pin 87 may include a stem and a roller to reduce friction between pin 87 and vertical surfaces 95 and 97 of slot 86. It should be noted that as the boom assembly 14 is pivoted about the boom pivot pin 20, the horizontal leg 63 of the support arm 62 is maintained in an absolutely horizontal position due to a parallelogram effect caused by the arrangement of the connection thereof to the boom base section 16 and the shipper link 67, both of which are pivotally connected to the shipper 18. Similarly, linkage arms 72 and 74, leg 63 and the horizontal portion 80 of cam 82 form a parallelogram to maintain the orientation of cam 82 during elevation of boom assembly 14 about pivot pin 20.

Pivotally mounted on the pivot pin 68 on the first vertical leg 64 of the support arm 62 is the lower end of a direct drive arm 89, the upper end of which is notched and adapted to abut a direct drive stop pin 88 fixed intermediate the length of the first parallel linkage arm 72.

The linkage movement modifying mechanism 48 comprises a drive link 90, the lower end of which is pivotally mounted on the pivot pin 70 pivotally supporting the second vertical leg 66 of the support arm 62. Mounted intermediate the length of the drive link 90 by a pivot pin 92, is a movement modifying link 94, the lower end of which is pivotally connected to the lower end of a motion modifying arm 96, pivotally supported on the pivot pin 68. The upper end of the arm 96 is notched and is adapted to abut a reverse motion stop pin 98 mounted intermediate the length of the direct drive arm 89.

The boom extension responsive assembly 50 comprises a spring motor 104 operatively connected to a cable drum 106 to urge reeling in of a cable 108. The free end of the cable 108 is connected to the fixed sheave block 28 of the boom assembly 14. The cable drum 106 is further provided with a drive shaft 110 on which is positioned a bevel gear 112 drivingly engaging a pinion 114 mounted on one end of a threaded traveling block support shaft 116. The shaft 116 is rotatably supported parallel to the longitudinal axis of the boom base section 16 by first and second pillow blocks 118 and 119. A threaded traveling block 120 is moveably mounted on the threaded portion of the shaft 116 and supports pin 87 projecting laterally therefrom and cam roller 121 rotatably supported on the upper end thereof. The cam roller 121 supports a cam bar 122 an inner end of which is pivotally mounted on the upper end of the first pillow block 118. One end of a flexible push-pull cable 124 is connected to the cam bar 122, the other end of the cable 124 being supported on the second parallel linkage arm 74 and connected to the upper end of the drive link 90. The cam bar 122 is adapted to adjust the position of the drive link 90 relative to the second parallel linkage arm 74 according to boom length, as shown in FIGS. 5 and 6. It should be noted that while the boom extension responsive assembly 50 is illustrated in FIG. 3 as being laterally displaced from the linkage assembly 46, in practice these assemblies are aligned with the pin 87 received between vertical surfaces 95 and 97 of slot 86, as indicated in phantom lines in FIG. 3. Alterations in the elevation of boom assembly 14 will cause pin 87 to slide between the vertical surfaces 95 and 97. It should be apparent that any appropriate method can be used to engage pin 87 for slideable movement along the horizontal leg 84 of cam 82.

The potentiometer assembly 42 is intended to provide an electrical resistance variable directly according to working radius as determined by boon angle and length. This assembly comprises first and second variable resistance potentiometers 142 and 144 including moveable first and second wipers 146 and 148, respectively, to adjust the resistance thereof. The wipers 146 and 148 are operatively adjusted by movement of first and second cam followers 150 and 152. An outrigger switch 154 which is adapted to be actuated by movement of the outriggers 36 to an operative position is provided to direct a signal from either the first or second potentiometers 142 and 144 to the indicator assembly 44.

The potentiometer 142 is adjusted by movement of the direct drive arm 89 which is connected by a first drive link 132 to a first crank arm 134. The first crank arm 134 is spring-loaded by a spring 135 to rotate in a clockwise direction, as shown in FIG. 3, and is fixed to a first horizontally disposed shaft 136 rotatably mounted on the boom base section 16. A first cam plate 138 is mounted on the shaft 136 so that lateral movement of the first drive link 132 will cause rotation of the first cam plate 138 and movement of the first cam follower 150.

The second potentiometer 144 is adjusted by movement of a second drive link 158 pivotally connected between the second parallel arm 74 and a second crank arm 160. The second crank arm 160 is fixed to a second horizontally disposed shaft 162 rotatably mounted on the boom base section 16. A second cam plate 164, which may be of a different shape than the first cam plate 138, is mounted on the shaft 162 so that lateral movement of the second drive link 158 will cause rotation of the second cam plate 164 and movement of the second cam follower 152.

It should be noted that while cam plates 138 and 164 may be used in conjunction with simple variable resistance potentiometers 142 and 144, in some instances it may be desirable to substitute variable tap or shape potentiometers for this arrangement thereby eliminating the need for the cam plates.

Operation of the overall device with the crane outriggers adjusted to support the crane, is as follows. Upon loading of the boom assembly **14**, the strain gage assembly **40** reads the strain of the lower cylinder pivot pin **23** and directs an electric signal through the amplifier assembly **41** to the indicator assembly **44**. The magnitude of the signal from the strain gage assembly **40** is proportional to the magnitude of the total load moment exerted about the boom pivot pin **20**. This signal is compared to a signal from the first potentiometer **142** which is directed through the switch **154** and which varies according to working radius. When the boom assembly **14** is positioned in a lowermost and fully extended position, the boom orientation responsive linkage assembly **46** is positioned as shown in FIG. **3**. Maximum permissible load moment, as indicated by segment A of curve B in FIG. **2**, is a minimum. Upon upward pivoting or elevation of the boom assembly **14**, the support arm **62** pivots about the pivot pin **70** but is maintained in an absolutely horizontal condition by virtue of the parallelogram effect induced by its connection to the boom assembly **14**, the shipper link **67** and the shipper **18**. Such pivotal movement between the linkage assembly **46** and the boom assembly **14** causes the pin **87** to move in an arc relative to point **70**. Pin **87** slides along at least one of the vertical surfaces **95** and **97** in slot **86** causing cam **82** to respond to the horizontal component of movement of pin **87**. The movement of cam **82** causes pivoting of the parallel linkage arms **72** and **74** in a clockwise direction, as shown in FIG. **3**, towards the position shown in FIG. **4**. Such arm movement permits lateral movement of the springloaded drive link **132** thereby changing the resistance of the potentiometer **142** an amount proportional to the additional load moment that may be safely carried because of a reduction in the moment arm.

Retraction of the first or second extensible boom sections **24** and **26** permits the spring motor **104** to rotate the cable drum **106** to reel in the cable **108**. Rotation of the drum **106** causes rotation of the traveling block support shaft **116** thereby threading the traveling block **120** laterally along the shaft **116**. Lateral movement of the traveling block **120** causes further pivotal movement of the parallel linkage arms **72** and **74** by virtue of the driving abuttment of the sliding pin **87** within the vertical surfaces **95** and **97** of cam follower bar slot **86** and consequential potentiometer resistance change. Such movement continues during boom assembly elevation or retraction until the movement modifying stop pin **98** on the direct drive link **89** is pivoted in a clockwise direction to a point short of that shown in FIG. **4**, where it abuts the end of the movement modifying arm **96** which has been moved by the drive link **90**. The movement modifying arm **96** is moved in a counterclockwise direction according to boom retraction by virtue of its connection with the push-pull cable **124** and the cam bar **122**. Further upward pivotal movement or retraction of the boom causes the movement modifying arm **96** to move in a counterclockwise direction and to likewise carry the direct drive link **89**, thereby oppositely adjusting the potentiometer **142** to change resistance and decrease the maximum permissible load moment setting.

When the crane **10** is operated without the stabilizing effect provided by the outriggers, substantially different maximum permissible load moment characteristics are entailed. A typical plotting of maximum permissible load moment when a crane is "operated on rubber" is presented by curve H in FIG. **2**. It will be seen that maximum load moments for corresponding operating ranges are substantially less than those of a crane stabilized by the outriggers. To compensate for this reduction, the outrigger switch **154** is moved to direct a signal from the second potentiometer **144** to the indicator assembly **44**. The second cam plate **164** is shaped so that elevation or retraction of the boom assembly **14** and consequential pivoting of the parallel arms **72** and **74** will cause appropriate movement of the second drive link **158** and the second wiper **148** to reflect the maximum permissible load moment indicated by curve H in FIG. **2**.

It should be noted that drive link **90** may be designed to either merely interrupt or reverse movement of the direct drive link at any desired rate during further upward movement or retraction of the boom assembly. In any case, the cut-off or reversing point, which varies as to boom length, may be accurately established to distinguish between the operation range wherein stability is critical as opposed to the operation range wherein structural strength is critical and where maximum permissible load moment is substantially reduced. Similarly, the potentiometer drive cam plates **138** and **140** may be designed to provide any desired rate of change characteristics.

Although the present invention has been described as a safe load indicator, it should be apparent that the basic concept thereof is applicable to a crane boom movement control environment wherein warning signals are supplemented or replaced by control features.

Further, various novel aspects of the present invention may be advantageously included in a crane boom position device wherein conventional means are provided to indicate control or boom orientation.

Finally, although but one embodiment of the present invention has been disclosed, it should be appreciated that various mechanical modifications or substitutions may be made, particularly of the drive and linkage arrangements, without departing from the basic inventive concepts to be ascertained from the following claims.

We claim:

1. In a boom orientation responsive device for a material handling apparatus including a boom mounted on a vehicle for pivotal movement about a horizontal axis, the vehicle adapted to be supported on a supporting surface, the boom adapted to support a load on the free end thereof at a plurality of positions having a plurality of moment arms about the horizontal axis, the device including boom orientation means for providing a signal responsive to a change in said moment arms, moveable means operatively connected to said boom orientation means to actuate the same, said moveable means being moveable in a first direction in accordance with a change in the length of the moment arm about the horizontal axis during movement of the boom in a first direction;

the improvement comprising:

movement modifying means for modifying the movement of said moveable means during further movement of the boom in the first

direction to alter actuation characteristics of the boom orientation responsive device wherein said boom orientation means provides a signal responsive to movement of the boom about the horizontal axis, the boom further being longitudinally extensible and said device further includes boom extension responsive means responsive to longitudinal extension or retraction of the boom for moving said moveable means and said movement modifying means and wherein said boom extension responsive means is connected to said movement modifying means for altering the point at which said movement modifying means modifies movement of said moveable means in accordance with boom length, said device further including load gage means for gaging the quantity of the load supported by the boom, said load gage means cooperating with said boom orientation means to provide a signal indicative of boom orientation and load, said device further including warning means connected to said boom orientation means and said load gage means, said warning means providing an indication of overload on the apparatus, wherein said load gage means includes strain gage means for measuring total moment about the horizontal axis about which the boom is pivoted, said boom orientation means including potentiometer means cooperating with said strain gage means and said moveable means.

2. In a boom orientation device according to claim 1 wherein said moveable means includes first eccentric means for varying the rate of adjustment of said potentiometer means per unit of boom movement and second eccentric means for varying the rate of movement of said movement modifying means.

3. In a boom orientation device according to claim 1 wherein said potentiometer means includes a multiple tap, shaped potentiometer.

4. In a boom orientation responsive device according to claim 2, wherein the vehicle comprises a self-propelled rubber-tired supported truck including moveable outrigger means for stabilizing the same, the device further including second moveable means operatively connected to said boom orientation means and moveable in a first direction in accordance with a change in the length of the moment arm about said horizontal axis during movement of the boom in a first direction, said second moveable means including third eccentric means for varying the rate of adjustment of said potentiometer means per unit of boom movement upon movement of the outrigger means to an inoperative position.

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