Aug. 2, 1966 J. ADAMSKI 3,263,515

FORCE TRANSFER SYSTEM

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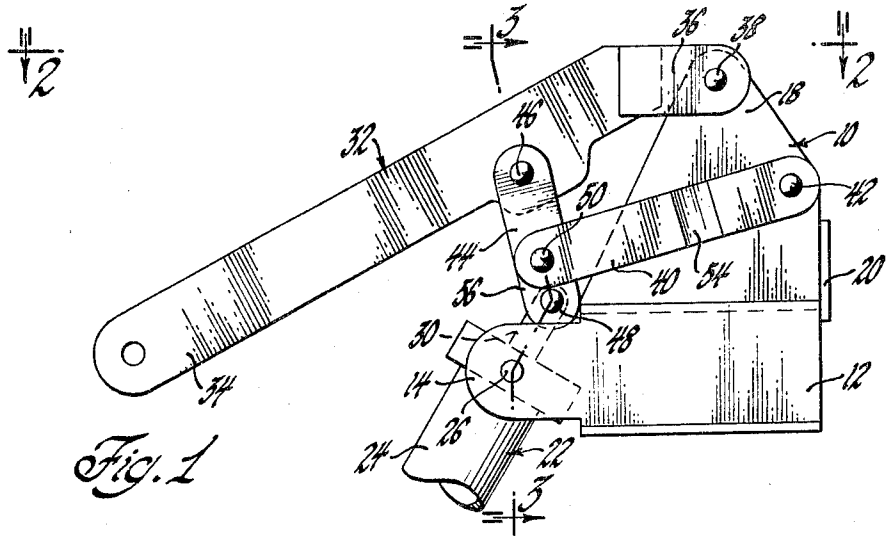


Fig. 1

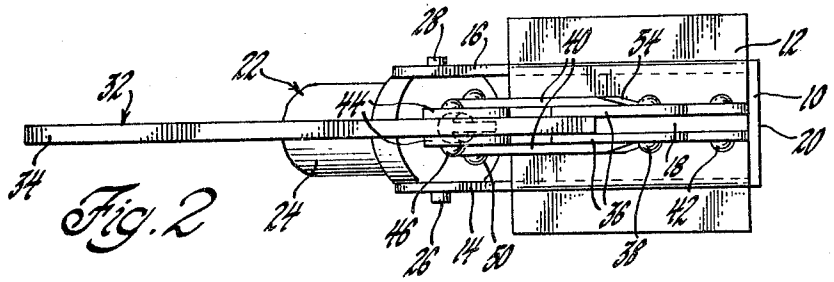


Fig. 2

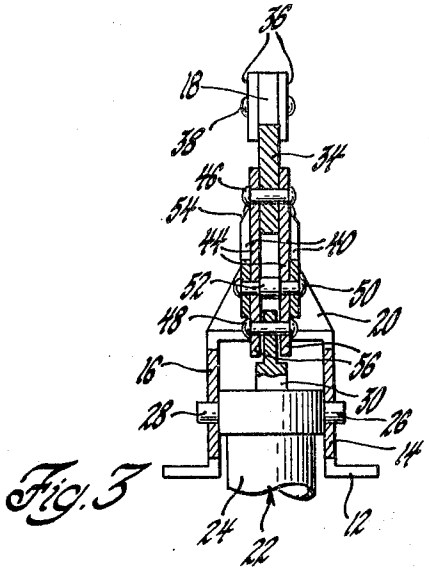


Fig. 3

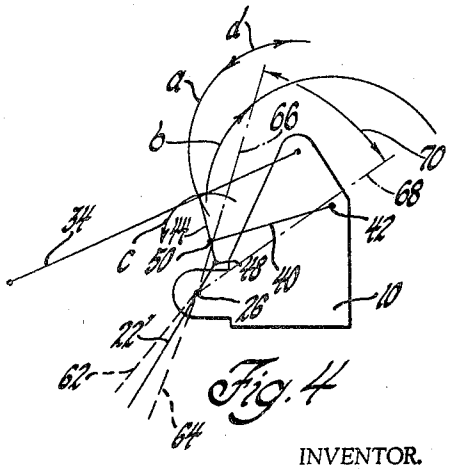


Fig. 4

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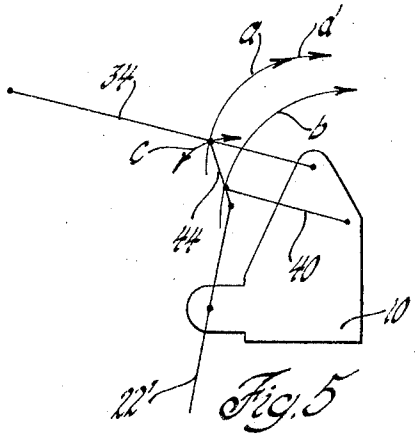


Fig. 5

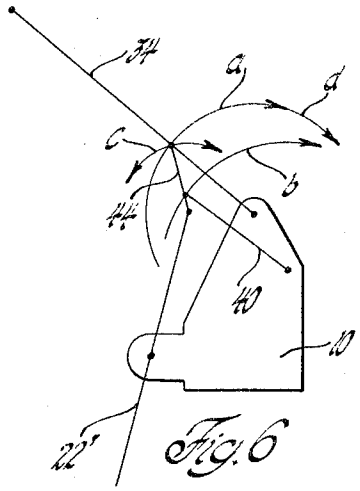


Fig. 6

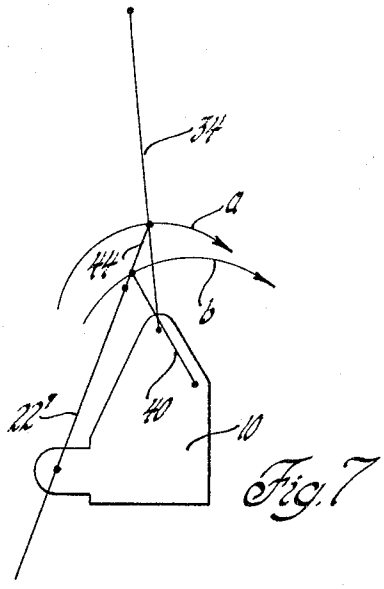


Fig. 7

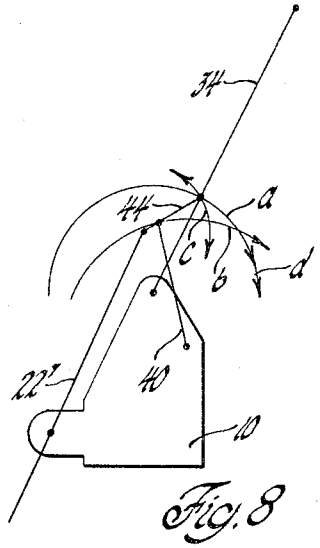


Fig. 8

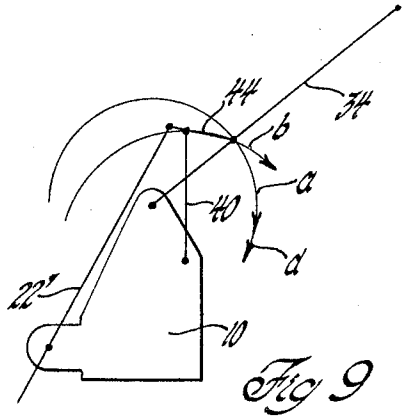


Fig. 9

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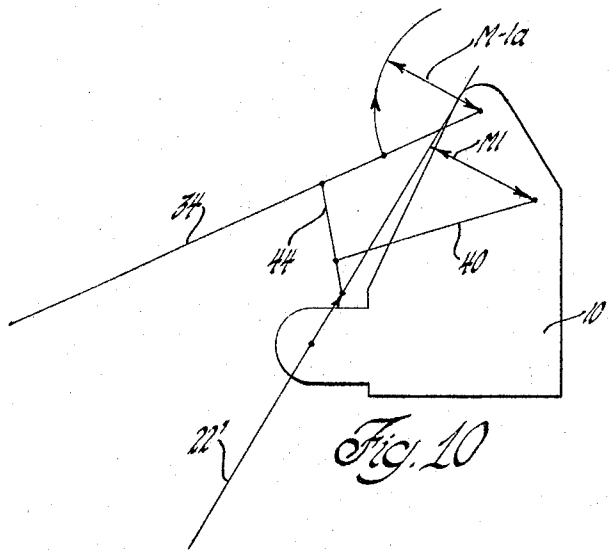


Fig. 10

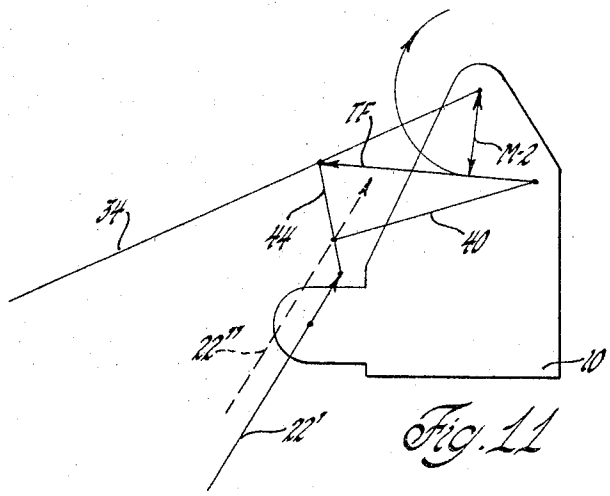


Fig. 11

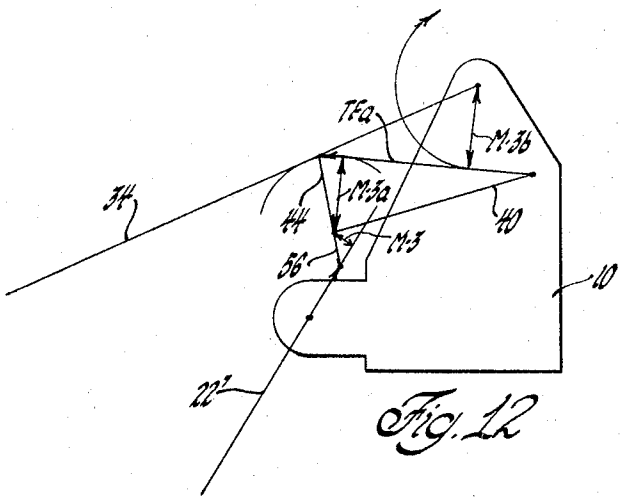


Fig. 12

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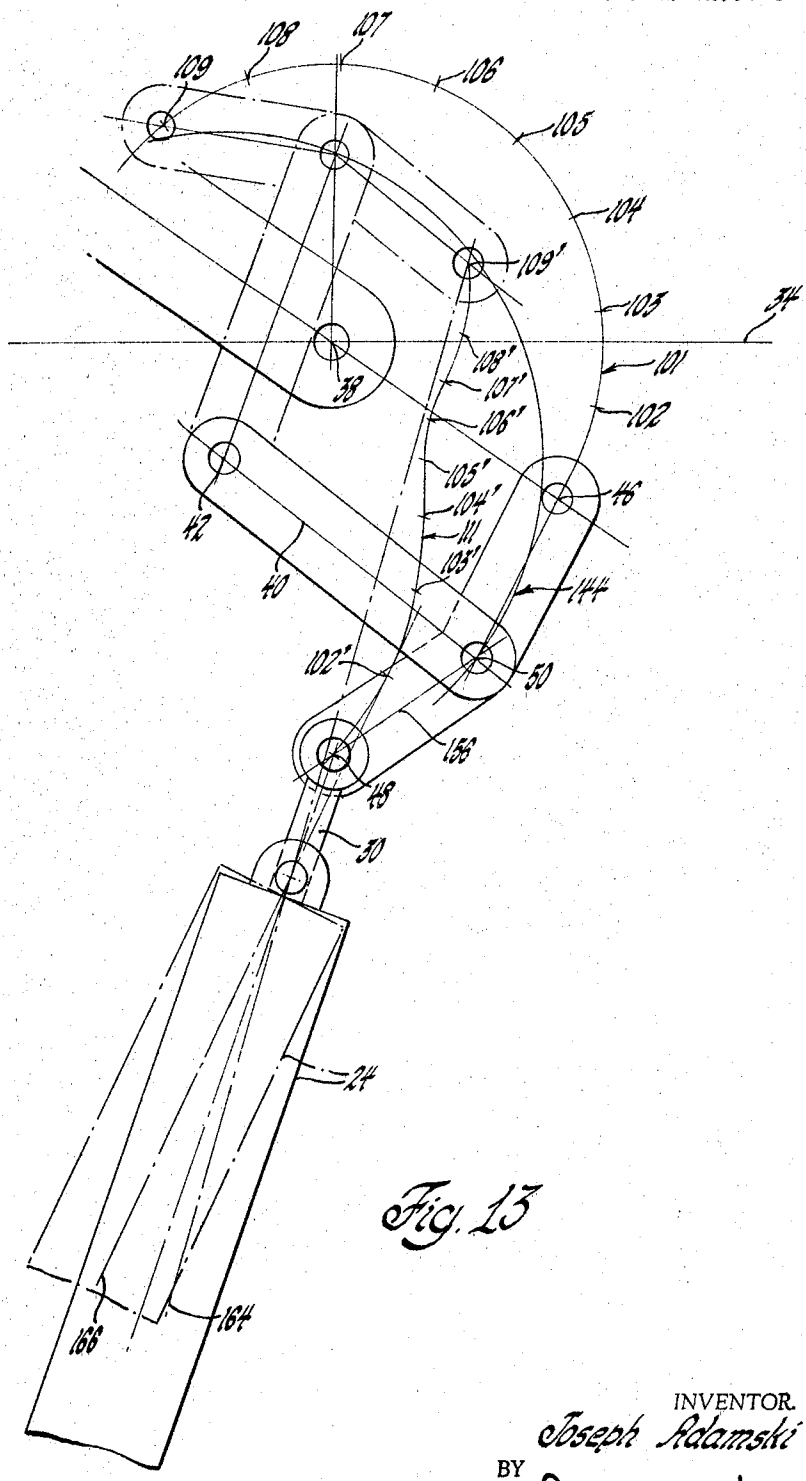


Fig. 13

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3,263,515
FORCE TRANSFER SYSTEM

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3 Claims. (Cl. 74—105)

This invention relates to power trains and force transfer systems in general and more particularly to new and novel means of using a linear force applicator to obtain more effective and efficient actuation of a rotary member.

There are numerous instances in which it is necessary or desirable to use a linear force applicator to obtain rotary or oscillatory movement of another member. Normally this is accomplished by connecting the linear force applicator directly to the rotary member at a point thereon which is spaced from its rotary or pivotal axis. This, in turn, requires that the point of application move with the rotary member and there is a consequent variance in the effective moment arm causing rotary movement as the linear force travels towards and away from the rotary axis of the driven member.

It will be appreciated that the effectiveness of the linear force is minimized as it approaches the axis of the rotary member and is completely lost at the commonly known dead-center position where it passes through the rotary axis. Accordingly, full rotary actuation by a linear force applicator normally requires some means, such as a fly wheel, to carry the rotary member through the dead-center position. For this reason pivotal or rocker arm actuation of a rotary member is normally limited to less than 180° to avoid such dead-center problems. However, the problem of lost power as the linear force approaches the dead center position still prevails.

In general, the linear force applicator is normally used to obtain pivotal or rocker arm movement of a rotary member only when the movement required is less than 180°. Even such a half-cycle is limited to those instances in which maximum power is required at the mid-cycle rotary position. Where maximum power is required to be applied at either or both of the forward and return actuating positions, as in a rocker arm arrangement for example, the effective use of the linear force applicator is limited to a travel arc of about 90°, beyond which limitation such linear force becomes progressively less effective upon a rotary member to a zero factor at the dead center position previously mentioned.

The operating mechanism of a folding top for convertible automotive vehicles provides a good example of the use of a linear-to-rotary transfer device wherein considerable improvement is needed. Some such installations require rotary movement of a top operating link, which is usually a rear side frame rail of the top structure, through an arc close to, if not exceeding, 180°. Furthermore, full power is needed to lift the top structure out of the storage well when erecting the top and off the windshield header bar in retracting and collapsing the top for storage. Sufficient power is also needed at both ends of the travel arc to hold the top structure erected and seated on the windshield header bar as well as to hold the retracted top structure securely seated and stacked in the storage well.

Most folding tops for convertible vehicles are operated by hydraulic power cylinders. Such linear force applicators are usually either pivotally mounted on the vehicle floor or trunnion mounted on a support under the rear side rail and are connected directly to such rear wide frame rail or a like member of the top operating structure. In general, different crank arm arrangements, variations in the interconnection of operative members, power transfer connections and links and the like have been used to obtain better operational efficiency. However, few innovations have been suggested as regards the disposition of the power cylinder or other linear force applicator. The prime concern in locating the power cylinder or other force applicator has been to obtain tangential travel of the applied force relative to the rotary axis of the driven member of the operating structure. As a consequence, the changes have been mostly in operating structure and have included longer cylinders to obtain a greater stroke, have required undue space to swing the cylinders, and have resulted in the exposure of operative parts, connecting arms, etc. which should not be visible.

One of the better folding top operating mechanisms makes use of parallel or trapezium type of four-bar linkage wherein the rear frame rail and balance links serve as crank arms and a power transfer link or like member serves as a connecting link therebetween. Numerous advantages are obtained in such an arrangement but still further improvement is obtainable therein in the application of the teachings of this invention thereto. For emphasis in this regard, reference will be made to the power transfer device of this invention as applied to such use in the subsequent discussion. However, it should be remembered that many other uses of equal importance may be made of the device disclosed.

It is an object of this invention to provide a power train or transfer system wherein a linear force applicator may be used to greater advantage to obtain rotary motion of a driven member.

It is an object of this invention to provide a linear-to-rotary drive system wherein arcs of 180° or more may be traversed by rotary members without dead center problems or appreciable power losses.

It is an object of this invention to provide a linear-to-rotary drive power train wherein power output of the rotary driven member may be eccentuated at end stroke positions and beyond the normal quadrant restrictions.

It is an object of this invention to provide a compact and simple operative mechanism for the transportation of linear forces to rotary actuating forces without minimization thereof and, in fact, greater and more effective utilization of the available power for the task required.

More specifically, it is an object of this invention to utilize a simple four-bar linkage system wherein one of the cranks is the rotary member to be driven and a drive force applied through an extension of the connecting link, between the customary driving and driven links, will obtain a leverage advantage and additional power in the course of rotary actuation induced by a linear force applicator.

Other objects of this invention include providing a power transfer device which is complete, simple and compact in arrangement, wherein movement arm forces may be obtained beyond normal over-center positions, wherein controlled force variation is prevalent throughout the entire operative cycle, and wherein the linear force applicator may be disposed essentially stationary.

These and other objects and advantages to be gained in the practice of this invention will be better understood and appreciated upon a reading of the following specification in regard to a preferred embodiment of the invention and having reference to the accompanying drawings wherein:

FIGURE 1 is a side view of a power transfer device incorporating the teachings of this invention.

FIGURE 2 is a plan view of the power transfer device shown in FIGURE 1 a sseen in the plane of line 2—2 therein.

FIGURE 3 is a rear and partially cross-sectioned view of the power transfer device shown by FIGURE 1 as seen

in the broken plane of line 3—3 therein and looking in the direction of the arrows.

FIGURES 4–9 are schematic sketches of the force transfer device on the other drawing figures showing progressive positions of the components thereof and used for force analysis purposes.

FIGURES 10–12 are also schematic sketches of the force transfer device of FIGURES 1–3 but are here used to show for force analysis of the different and cumulative forces derived from the disclosed system.

FIGURE 13 is a partial schematic of a modified force transfer device having the travel of the power actuator shown graphically thereon.

The power transfer device shown by the drawings includes a mounting bracket **10** having a base portion **12** of inverted channel cross-section. The side walls of the base are extended at one end to provide trunnion supports **14** and **16** and a support plate **18** is disposed in a vertically upright position over the base **12**. A triangular plate **20** is secured to the back of the base **12** and support plate **18** to provide vertical rigidity for the latter.

The linear force applicator is a power cylinder **22** of the hydraulic fluid type. The cylinder portion **24** is disposed between the trunnion supports **14** and **16** and mounted on the trunnion support pins **26** and **28**. A piston rod **30** extends from the trunnion supported end of the cylinder in the plane of the support plate **18**.

An operative four-bar linkage **32** is provided on the mounting bracket **10**. It includes a driven member **34** which serves as one of the cranks and might be such as the rear frame side rail member of a folding convertible top structure.

Driven member **34** has a forked end **36** and is secured to the upper disposed end of the support plate **18** by a pivot pin **38**. The driven member is accordingly rotatable essentially in the plane of the support plate.

A control link **40**, which is actually a pair of links disposed in parallel spaced relation, has one end secured to the support plate **18** on a pivot pin **42**. The control link rotates about the axis of the pivot pin **42** in unison which the driven member **34** in a manner to be presently described.

A connecting link, **44**, which may subsequently be referred to as a thrust transfer or rocker link, is shown as a pair of parallel spaced links secured by a pivot pin **46** to the driven member **34** and having the other ends thereof secured by a pivot pin connection **48** to the extended end of the piston rod **30**.

The control links **40** have their extended ends connected by a pivot pin **50** to the connecting links **44** intermediate the ends thereof. A spacer **52** is provided between the parallel spaced connecting links **44** and a common pivot pin **50** is used. It will also be noted that the control links **40** are formed to include an offset **54** intermediate their ends for more suitable engagement with the support plate **18** and with the connecting links **44** disposed in different reference planes.

It will be appreciated that the effective length of the connecting link **44**, as such, is that portion of the links between the pivot pins **46** and **50**. The extended end **56** of the connecting link, have the lineal force actuator connected thereto, causes it to also serve as a rocker or lever as fulcrumed on the control links **40**.

Referring now to FIGURES 4–9:

In the schematic sketches only the mounting bracket **10**, driven member **34**, control link **40** and connecting link **44** are specifically identified. The line of linear force application has been identified **22'** for reference back to the use of the hydraulic power cylinder **22** for such purposes. In all other respects reference should be made to FIGURES 1–3 for structural details mentioned in the subsequent discussion.

With the driven member **34** disposed as shown by FIGURE 4, the line of linear force application **22'** introduces an operative force into the operative linkage through the end of the connecting links **44**. This force is, in part, a compressive thrust force which is transmitted through the connecting link to the driven member **34** for rotary actuation thereof about its fixed axis to the support plate **18**.

At the same time, the control link **40** is similarly receptive of the moment arm force through the connecting link **44** for rotary actuation thereof about its fixed pivotal connection to the support plate **18**.

In addition, the connecting link **44** itself is receptive of a moment arm component tending to induce counterclockwise rotation thereof about its pivotal connection to the end of the control link **40**. This, in turn, introduces a tensioning component in the driven member **34** and a force component normal thereto which is additive to the first mentioned rotary drive force and thereby provides accentuated rotary actuation of the driven member **34**.

Considering the application of force to the operative mechanism in another manner, the connecting link **44** and control link **40** may be considered parts of a toggle link joint operative of the driven member **34**. If the applied force were at the pivotal connection of the connecting link **44** and control link **40**, a rotary drive, as shown by the arrows *a* and *b*, would normally result as the joint was spread and cranks were induced to move in a clockwies direction about their fixed pivotal axes. With the applied force provided on an extended end of the connecting link **44**, a counter rotating force, as shown by the arrow *c*, is introduced and a leverage accentuated rotary drive is introduced into the driven member **34**, as shown by the arrow *d*, in the further spread of the toggle link joint.

Reference of FIGURES 10–12 where the forces are shown separately will help to understand what happens a little better.

In FIGURE 10 the linear force **22'** is shown to have a moment arm **M–1** with respect to the link **40**. The resultant force moment acts through the operative linkage to rotate the driven link **34** in a clockwise direction the same as if the moment arm was **M–1***a*.

FIGURE 11 shows the linear force **22'** offset to the toggle joint pivot **50** (of links **40** and **44**) and identified as **22''**. At such point it causes the toggle spreading force identified by the arrow TF; the link **40** being fixed to the mounting bracket **10**. This, in turn, produces a moment arm **M–2** with respect to the driven link **34** and another clockwise rotating force with respect thereto.

FIGURE 12 shows the linear force **22'** as producing a moment arm **M–3** with respect to the short end **56** of the link **44**. This is transposed to a moment arm **M–3***a* at the longer end of link **44** and with respect to the toggle spreading force TF*a* created by the rocker arm action of the link **44**. The moment arm **M–3***b* results with respect to the driven link **34** and a still further clockwise rotating force is imposed thereon.

Returning now to the operative sequence following from the application of power to the driven link **34** through the link **44**:

As the driven member **34** attains the radial position shown by FIGURE 5, the thrust force component through the thrust or connecting link **44** is greater in that the line of linear force application **22'** is in closer alignment with the thrust or connecting link. The application of greater force is also apparent in that the line of linear force application **22'** is spaced further from the fixed pivotal connections of the driven member **34** and the control link **40** than in FIGURE 4 and the moment arm forces will accordingly be greater.

At the same time, the counter rotating force on the connecting link **44** is less; again, due to the closer alignment of the line of linear force application **22'** to the connecting link **44**. In the line of linear force application **22'** passes closer to the pivotal connection between the connecting link **44** and the control link **40** than was

the situation in the arrangement of FIGURE 4, and the counter rotating moment arm force is accordingly less.

It will be appreciated that while the rocker or leverage accentuated rotary drive force is less between the linkage positions of FIGURES 4 and 5, the thrust force through the link **44** has increased. This rocker or leverage drive force persists throughout the operation of the linkage (except in one instance which will be identified later) and may be varied by different proportioning of the link or offset shapes thereof. Certain aspects of this invention in this regard will be further appreciated in the subsequent discussion and in the description of the modification shown by FIGURE 13.

Referring now to FIGURE 6, it will be noted that the line of linear force application **22'** is approaching alignment with the connecting link **44**. Referring back to FIGURES 4 and 5 and forward to FIGURE 7, it will be appreciated that the connecting link **44** is actually rotating in a clockwise direction about its pivotal connection on the end of the control link **40** rather than in the counterclockwise direction as might be expected by the application of the lineal force to the extended end thereof. This is due to the racing pivotal centers in the rotary actuated members and the fact that the counter-rotational force is dissipated in the spreading of the toggle linkage as described earlier.

This additive aspect of the leverage induced spreading force on the toggle linkage, and through to the driven member **32**, continues until the link **34** is aligned with the fixed pivot of the link **40**. This closely approaches the position at which the linear force **22'** comes into alignment with the connecting link **44**, as shown by FIGURE 7, and can in fact by design be made to coincide therewith.

The length of the control link **40** and of the longer end of the connecting link **44**, that is the part between the links **44** and **32**, is purposely made greater than the distance between the fixed pivot point **42** of the control link on the mounting bracket **10** and the pivotal connection **46** where the connecting link **44** joins the driven link **32**. This is so the toggle link is never lost. The action is one of first spreading and then collapsing the toggle link, as will be subsequently shown, with a leverage accentuated drive obtained in each instance.

Referring now to FIGURE 7, it will be noted that the line of linear force application **22'** has come into alignment with the connecting link **44**. In such a situation there is no leverage force applied through the connecting link **44** but merely a rotary motion inducing thrust force applied therethrough. Although the rotary force moments are less than heretofore or subsequently, due to the closer disposition of the line of linear force application **22'** to the pivotal axis of the driven member **34**, this is of no consequence in an operating mechanism for folding convertible tops since, at such moment, the folding top mechanism is disposed essentially in a vertical and balanced position where minimal forces are required to carry it over center.

FIGURE 8 shows that the leverage forces are again introduced through the connecting link **44** as the driven member **34** passes beyond its over-center position. The rotary inducing thrust force through the connecting link **44**, and affecting the control link **40**, prevails and a clockwise rotary moment is induced in the connecting link **44** with a consequent leverage accentuated rotary drive of the driven member **34** as shown by the arrow *c* and the additive component arrow *d*.

FIGURE 9 shows the end position of the driven member **34** and the forces applied thereto. It will be noted that these forces are considerably greater than would otherwise be obtained in driving through the pivotal connection of the control link **40** with the connecting link **44**. In fact, this would be virtually impossible since it would require that the line of linear force application **22'** pass through the support plate **18** and so close to the pivotal axis of the supporting member as to approach a dead-center position. The leverage advantage obtained is most clearly and emphatically emphasized in the arrangement of parts shown by FIGURE 9.

Reference back to FIGURE 4, and in particular to the phantom lines **62** and **64** on each side of the line **22'**, representing the linear force applicator, shows that the force applicator has a very small amount of arcuate travel despite the broad sweeping arc which it imposes on the driven member or link **34**.

FIGURE 4 is also of further significance in appreciating that the line of linear force application **22'** should fall between the pivotal connections **50** and **42**, as shown by the dotted lines **66** and **68** and represented by the arc **70**, to obtain full advantage of the rocker arm force of link **44** and not have it work to the disadvantage of the system. Should the applied force be to the left of the pivot **50** the effect would be to collapse rather than spread the toggle linkage. Should it be to the right of the pivot **42** of link **40** on the mounting bracket it would obviously not obtain the desired rotational force on the control link. Preferably the force should be applied between the pivot point **38** of the link **32** on the mounting bracket **10** and the pivot point **50**. However, the best balance between the leverage accentuated drive and thrust drive through the connecting link **44** is considered best obtained in the arrangement first mentioned.

A fuller appreciation of the application of greater power at the ends of the power stroke of the linear force applicator, and of the relatively small arc of pivotal movement required thereby to affect a large arcuate travel of a driven member, will be had by reference to FIGURE 13.

Essentially the same linkage arrangement is provided as that which was previously described. Accordingly, like reference numerals are used to identify like parts and to avoid the necessity of re-describing the basic system. The principal change is in the use of a bell crank connecting link **144** or, more properly described, a connecting link having the rocker end **156** offset inwardly with respect to the four-bar linkage **132** disposed thereover.

The use of an inwardly or outwardly offset end on the connecting link **44** is principally for positional advantage as regards the force applicator and to obtain a straight line power stroke.

The swinging motion of the power cylinder **24** is practically nil in the arrangement shown; the lines **162** and **164** representing the outer disposed axial positions assumed.

The arc of the driven link **32** is identified by the numeral **101** and it has been divided into several equal arcuate segments between its end positions with the intermediate positions identified at **102–109**, consecutively. The corresponding positions of the point of force application to the end of the connecting link **144** are identified **102'–109'**, consecutively. These points are connected by a curve **111** which shows the relatively straight line of travel of the piston rod **30**.

It will be noted that the variations in the increments of travel represented by the points **102'–109'** show a greater travel at the beginning and ends of the power stroke than in the intermediate range. This clearly emphasizes that a greater activating force is applied to the driven link **32** at the lift-off and set-down positions most advantageous in many applications and including the folding top uses mentioned previously.

It should be obvious that the linkage system disclosed lends itself to various uses and that different variations of the geometric patterns provided by the pivot points, within the teachings set forth, will enable a wide variation of force potentials to be obtained.

It will be appreciated that the rever situation prevails in the application of a retracting force to the end of the connecting links **44** or **144** and that the resultant forces are similarly but oppositely applied to the different components. Thus, as applied to a folding top for convertible

vehicles, the actuating forces at the ends of the arc of rotary movement will in both instances be the greatest.

The force transfer linkage of this invention can be seen to produce three different force moments which work together to produce the desired result. All conflict between such force moments, as are present in other mechanisms of like kind, has been avoided. The compounding of the three force moments and their application with respect to the driven rotary member **32** makes possible the use of a shorter stroke linear actuator and a linkage which is less massive and includes shorter operative links. The pivot geometry is extremely compact and the resultant mechanism is most efficient.

Without further discussion it should be obvious that numerous other variations, modifications and combinations are conceivable and within the scope of this invention.

Although a preferred embodiment of this invention has been specifically shown and described, with reference to a particular use, it will be appreciated that this has been done to illustrate the scope of the present invention and without intent to unnecessarily limit the invention in any regard. Accordingly, such improvements, modifications and alterations as are within the spirit of this invention and are not specifically excluded by the language of the hereinafter appended claims are to be considered as inclusive thereunder.

I claim:

1. Power actuating linkage means comprising a mounting plate, a main lever pivotally mounted adjacent one end to said plate, a control link, means pivotally mounting one end of the control link to said plate, a linear actuator, means pivotally mounting one portion of the linear actuator to said plate, a rocker link, means pivotally mounting one end of the rocker link to said main lever, means pivotally mounting the other end of the rocker link to another portion of the linear actuator, means pivotally mounting the other end of the control link to the rocker link intermediate the pivotal connections at the ends of the rocker link, the said pivotal connections being positioned such that the line of linear force application from the linear actuator to the rocker link is between the pivotal connections of the control link and the plate and the control link and the rocker link during the high torque portion of the duty cycle of the power actuating mechanism.

2. The invention defined in claim **1** wherein the linear actuator comprises a cylinder and piston unit.

3. The invention defined in claim **1** wherein said high torque portion of the duty cycle is at one end of the travel of the power actuating mechanism.

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