TOWPLANE HOOK DESIGN AND OPERATIONAL TESTS

W. J. Betts, Arlington, Va., U.S.A.
G. P. Layton, Jr., Edwards, Calif., U.S.A.

INTRODUCTION

In the past, a number of glider/towplane upsets have occurred. Typically, these upsets have started when the glider either pulled directly upward in straight flight or high and outside on a turn. These types of maneuvers have then exceeded the nose-up pitch control capability of the towplane, resulting soon thereafter in a near vertical attitude of the towplane from which recovery is impossible unless the towline is released. Post-flight discussions with the pilots have revealed that they made every effort to release the towline from the towplane but could not exert enough force on the mechanism to do so. In cases where recovery was affected, it was subsequently discovered that the towline broke after altitude losses exceeding 1000 ft.

As a result of the tow pilots' observations that the release mechanism could not be operated under load, a study was made of the characteristics of these mechanisms.

An analytical study of the tow hook was performed and several tow hook configurations were load tested as was a typical operating linkage in a tow aircraft.

ANALYTICAL STUDY

In the analysis, the following contributions to hook release load were included:

(a) hook stability
(b) friction.

The effect of the diameter of the latch pin was not included since, for most hooks, this contribution is an order of magnitude smaller than the other factors. To more easily separate variables, the release load was determined as a function of moment about the hook pivot, then this moment was determined as a function of tow tension and tow angle as shown in Fig. 1.

It was assumed that each component could be analyzed separately then added to obtain the total release load. Figure 1 is a sketch of the hook and defines the symbols used in the following analysis.

CONTRIBUTION OF HOOK STABILITY

As shown in Fig. 1, the angle of the hook arm with respect to the latch arm tends to return the latch arm to a closed position; hence, it is a stable configuration.

The magnitude of this force is determined by the force normal to the latch arm at the point where the latch pin rides on the hook arm. The normal force on the latch pin (N) is:

$$N = \frac{M_P}{b + c \operatorname{Sin} \alpha}$$

Resolving this force into components normal and parallel to the latch arm yield the force normal to the latch arm as:

$$N \operatorname{Cos} \beta \operatorname{Sin} (\alpha - \beta)$$

Then, taking moments about the latch arm pivot, the unlatching force due to stability is:

$$T_{2_{stability}} = \left(\frac{M_P\ C}{b + c \operatorname{Sin} \alpha}\right)\left(\frac{\operatorname{Cos} \beta}{d + c}\right)\left[\operatorname{Sin} (\alpha - \beta)\right]$$

CONTRIBUTION OF FRICTION

As shown in Fig. 1, the friction contribution is a function of the release latch angle. This friction is determined by the magnitude of the normal force (N) resolved parallel to the axis of the latch arm. This force is:

$$N \cos B \cos (\alpha - \beta)$$

Multiplying by the coefficient of friction yields the force normal to the latch arm at the point of contact of the latch pin with the hook arm. Taking moments about the latch arm pivot, the unlatching force due to friction is:

$$T_{2_{friction}} = \left(\frac{M_P\ C}{b + c \sin \alpha}\right)\left(\frac{\cos \beta}{d + c}\right)\left(f \cos (\alpha - \beta)\right)$$

Also, the contribution due to friction is a function of the coefficient of friction. A value of 0.75 was assumed for the static coefficient and a value of 0.40 was assumed for sliding friction.

Thus the total load is:

$$T_{2_{total}} = \left(\frac{M_P \cdot C}{b + c \sin \alpha}\right)\left(\frac{\cos \beta}{d + c}\right)\left(f \cos (\alpha - \beta) + \sin (\alpha - \beta)\right)$$

The determination of M_P as a function of γ and T, then, must be determined for each type of hook geometry.

LOAD TESTS

By means of a weight and fulcrum arrangement, the hooks in question were load tested at loads from zero to 764 lb, and the release load measured. Due to the limitations of the test equipment, all loads were applied parallel to the axis of the hook, i.e., with no vertical- or side-force component.

Tests on Hooks

Four hooks were tested: the basic hook, Schweizer Part Number 12112-15 (Configuration A); a basic hook with a radius on the hook arm (Configuration B); a basic hook with a radius on the hook arm and a longer and articulated latch arm, arm (Configuration C), a longer latch arm, and a ball bearing race on the latch pin (Configuration D). These hooks are described in Figs. 2a and 2b.

Tests on Airplanes

In addition to loading the hooks, the installation in the aircraft from the hook to the cockpit was tested. In the installation tested, the mechanism consisted of a flexible push-pull cable (choke cable) from the cockpit back to a point four feet forward of the hook. At this point, a length of stranded aircraft cable was attached. This cable passed through a tube out the tail of the aircraft where it was terminated at the hook.

The release load at the cockpit was measured with a steady load applied at the latch arm.

COCKPIT ACTUATOR LOAD CRITERION

Based on discussions with tow pilots, it was determined that it must be possible to release the towline with a tension equal to the breaking strength of the towline applied at an angle of 30 deg. The breaking strength of towlines in common use average up to 900 lb.

From tests on typical "T" handles, it was determined that a practical maximum allowable release load at the cockpit actuator was 35 lb.

RESULTS AND DISCUSSION

Analytical Study

The equation developed in the analysis was applied to the operational tow release, Schweizer Part #12112-15, (Fig. 2

Configuration A). The physical characteristics of this hook and the variation of the dimension, a, with the tow angle α are shown in Fig. 3.

The variation of M_p with γ and T was first calculated and is shown in Fig. 4.

The variation of T_2 with M_p was calculated for the four cases listed below and is shown in Fig. 5:

Case I $\alpha = \alpha_0$, $f = 0.75$ (static)

Case II $\alpha = \alpha_0$, $f = 0.40$ (sliding)

Case III $\alpha = \alpha_f$, $f = 0.75$

Case IV $\alpha = \alpha_f$, $f = 0.40$

Case I then represents the load necessary to start unlatching the hook, and Case IV represents the most likely maximum unlatching load.

Hook Load Tests

The operational release (Configuration A) was tested and compared with the analysis. As shown in Fig. 6, the analytical result, when corrected for the zero load release force, compares reasonably well at the lower towline tensions, but deviates considerably at the larger towline tensions. Also shown in Fig. 6, the basic hook exceeds the 35-lb release criterion at a 435-lb towline load even without any further amplification of this load by the releaseable installation.

Configuration B was then tested to explain the difference between the analytical result and the test result for Configuration A. This configuration had the same mechanical advantage as Configuration A, but the hook arm was radiused so that the friction component was the only contribution to latch load. The results of these tests (Fig. 6) showed that a coefficient of sliding friction of 0.46 was required to match the initial slope. It is also likely that the assumption of a constant friction coefficient was in error, since the function is not a straight line.

A third configuration, C, was tested to determine the characteristics of another typical hook. This hook had a radiused hook arm and a latch arm with more mechanical advantage. The latch arm was also segmented in an attempt to provide a prying action on the hook arm. The characteristics of this hook are shown in Fig. 6.

A fourth hook, Configuration D, was built and tested to demonstrate the combined effect of radiusing the hook arm, increasing the mechanical advantage, and decreasing the friction by means of a ball bearing race on the latch pin. As shown in Fig. 6, these combined effects resulted in only a 5-lb latch load at a 750-lb towline load. Based on the hook geometry, the latch load for this hook would increase to 11 lb with a towline load of 900 lb applied with $\gamma = 30$ deg.

Aircraft Mechanism Tests

The results of the tests to determine the actuation lo-d at the cockpit as a function of the release load at the hook are shown in Fig. 7. This curve indicates that, even with no towline load applied, the 35-lb criterion would be exceeded.

Some preliminary tests with a series of pulleys indicated that the release load could be transferred from the hook to the cockpit with no more than a 10% increase, thus satisfying the 35-lb criterion with even Configuration B or C.

CONCLUSIONS

1. With the operational system tested, the tow pilot would not be able to release his hook at towline tensions exceeding 225 lb.

2. A reasonable maximum release load at the cockpit is 35 lb with a towline tension of 900 lb at an angle of 30 deg from the horizontal.

3. The basic hook with no allowance for the mechanism between the hook and the cockpit exceeds reasonable release load limits for towline tensions exceeding 435 lb.

4. The release load at the hook can be reduced by:

   (a) increasing the mechanical advantage

   (b) reducing friction

   (c) reducing the hook stability.

5. By the proper combination of relieving the hook release load and reducing the friction between the hook and the cockpit, the hook can be made to release with reasonable force.

TOW HOOK GEOMETRY

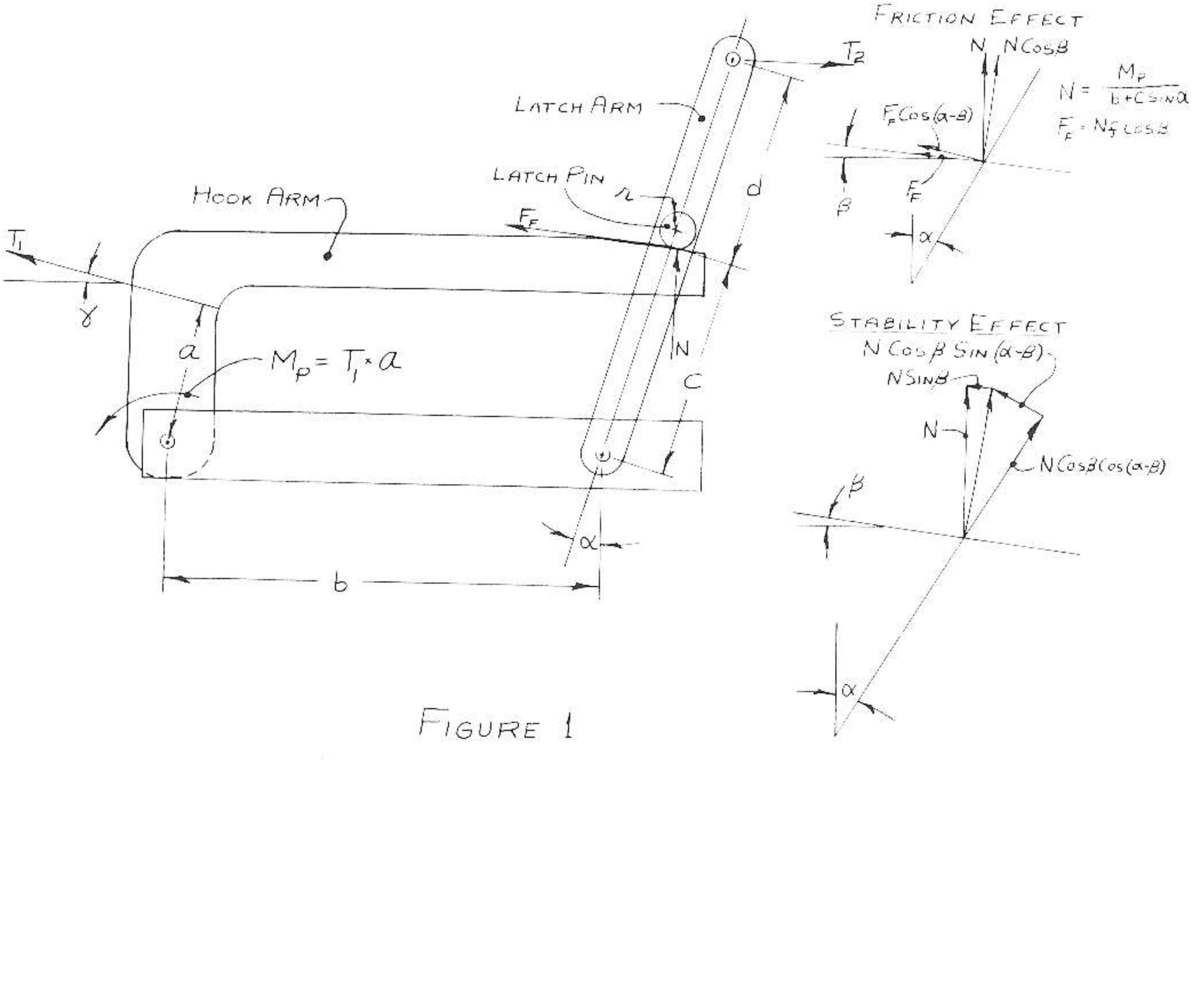


FIGURE 1

Figure 2b. Tow Hooks Tested--Physical Characteristics

		DIMENSION					
		b (in.)	C (in.)	d (in.)	α_0 (deg)	α_f (deg)	β (deg)
CONFIGURATION	A	1.75	1.02	0.75	11.5	30.5	11.5
	B	1.75	1.02	0.75	0.0	32.0	$\beta = \alpha$
	C	1.75	--	--	--	--	--
	D	1.75	1.02	1.40	0.0	32.0	$\beta = \alpha$

TOW VECTOR TO PIVOT DISTANCE
VS.
TOW ANGLE
(ALL CONFIGURATIONS)

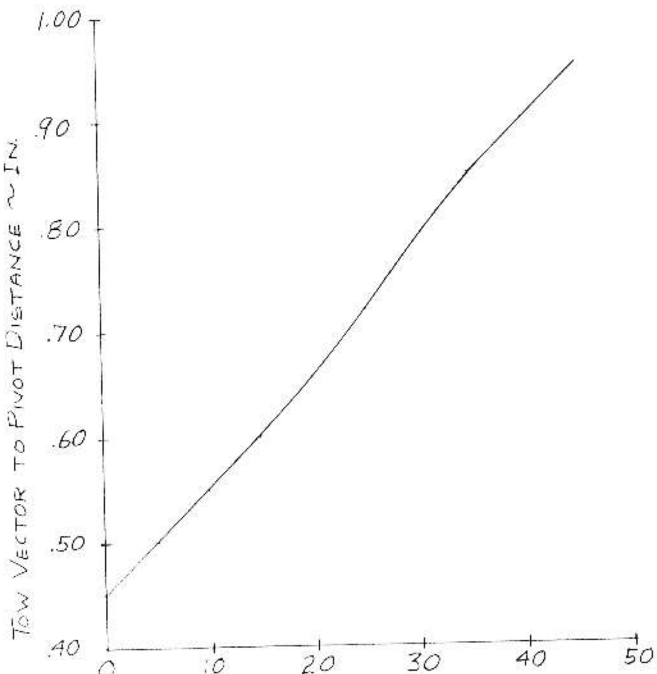


FIGURE 3

HOOK ARM MOMENT
VS
TOW TENSION
AND
TOW ANGLE
(ALL CONFIGURATIONS)

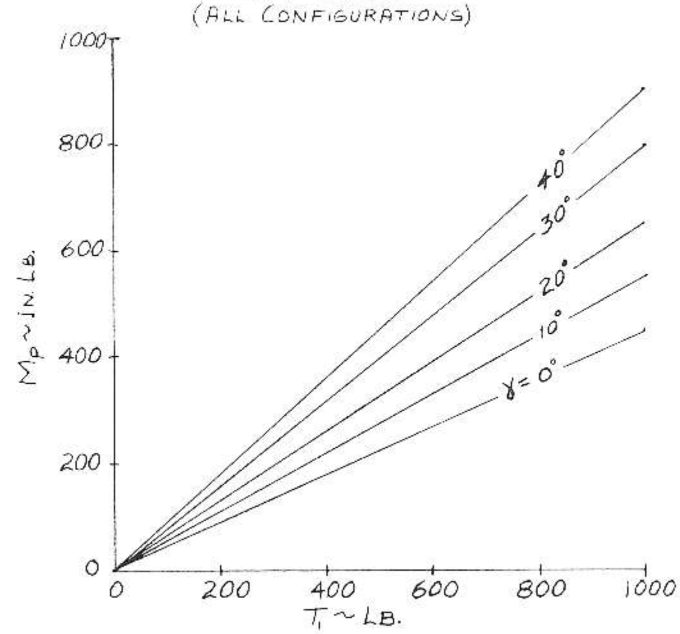


FIGURE 4

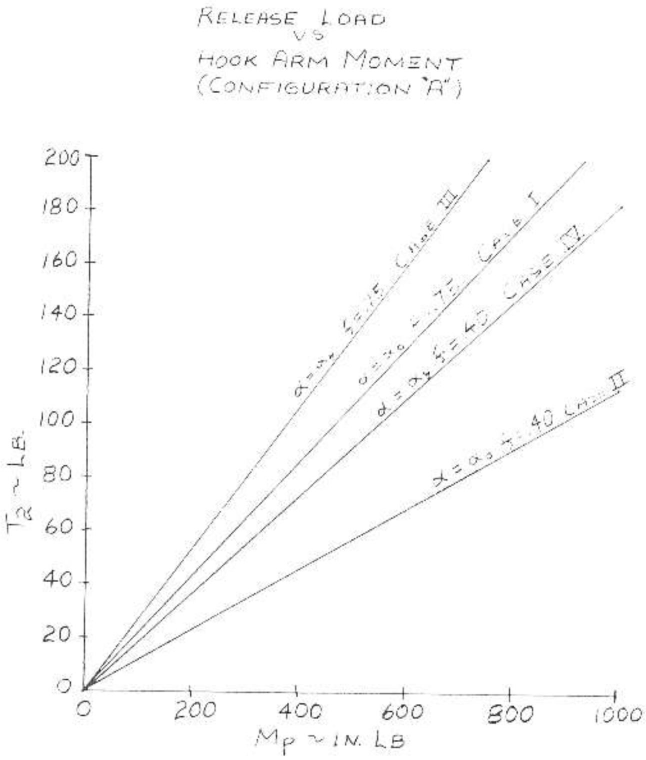


FIGURE 5

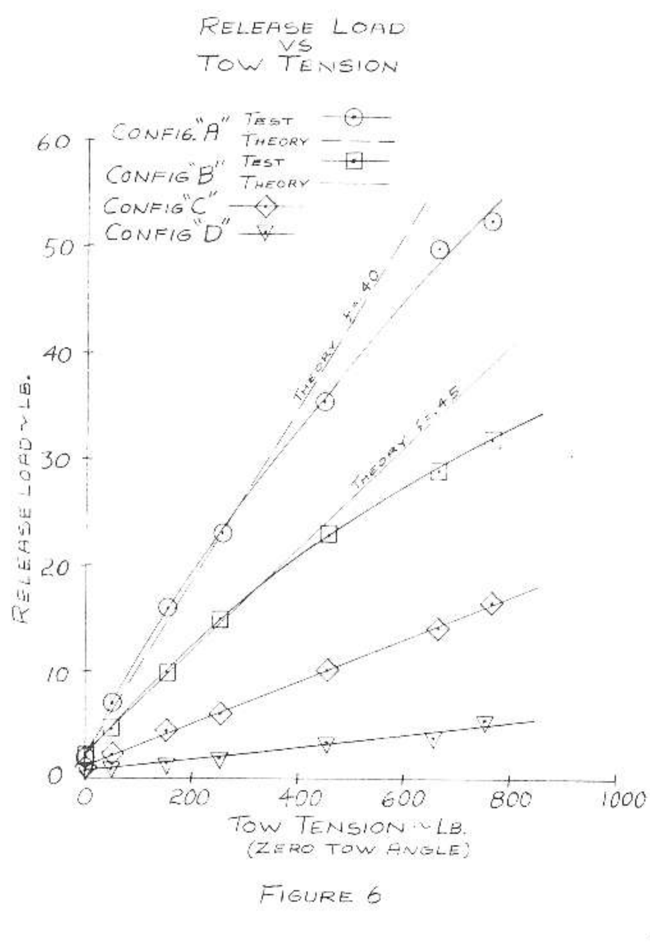


FIGURE 6

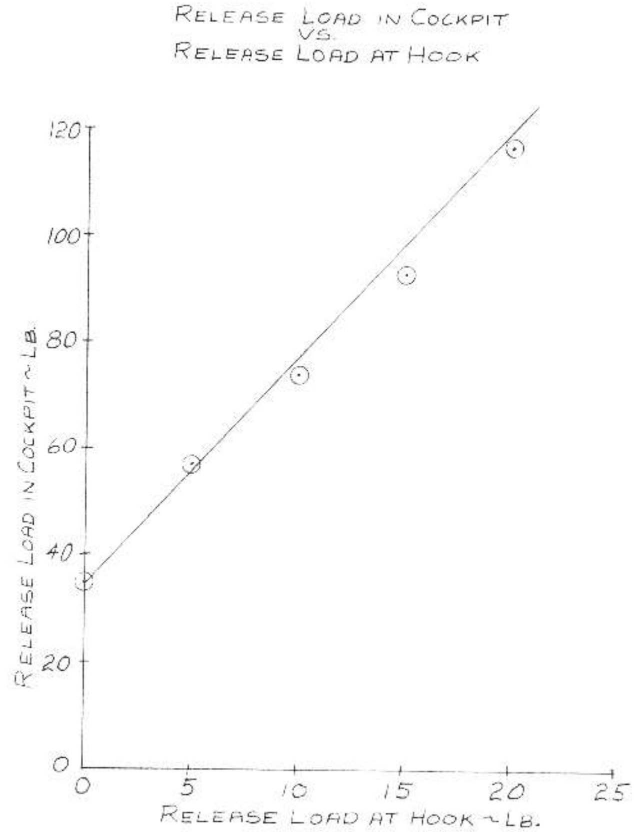


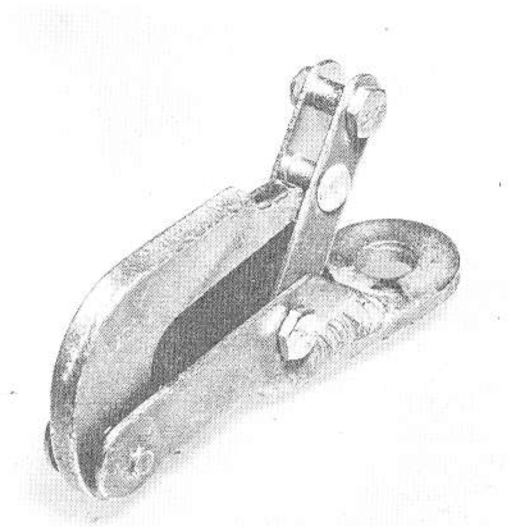
FIGURE 7

EDITOR'S NOTE

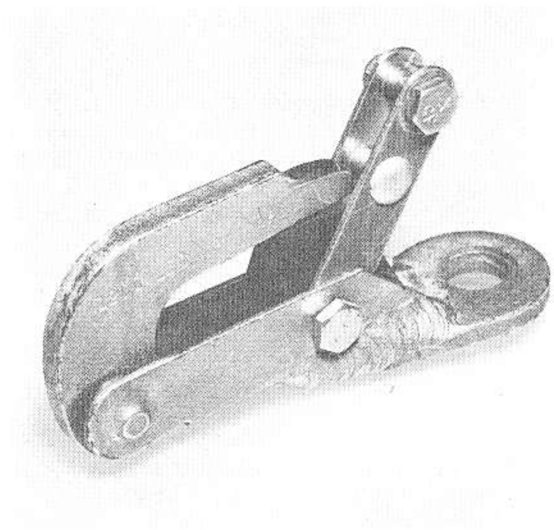
Schweizer Aircraft Corporation has pointed out to us that all production hooks are tested to release under 60 lb at 600 lb line pull using control lever 3414D. The mechanical advantage has been increased from about 1.9 to about 3.3 and no difficulty should be encountered in meeting the 35 lb maximum criteria. An adequate release control system should be used on existing hooks. All new installations should be tested with at least 600 lb on the tow line. Furthermore, periodic maintenance checks of the hook and release mechanism are very desirable.

The editors have observed that these hooks mounted on Cessna aircraft and retained with a single vertical pin have the possibility of rotating under side load and thereby greatly increasing the release load. We invite correspondence from readers who may have observed this in flight or ground tests.

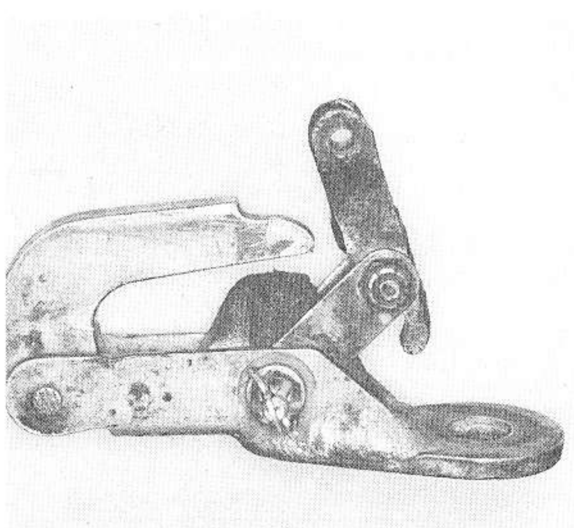
TOW HOOKS TESTED



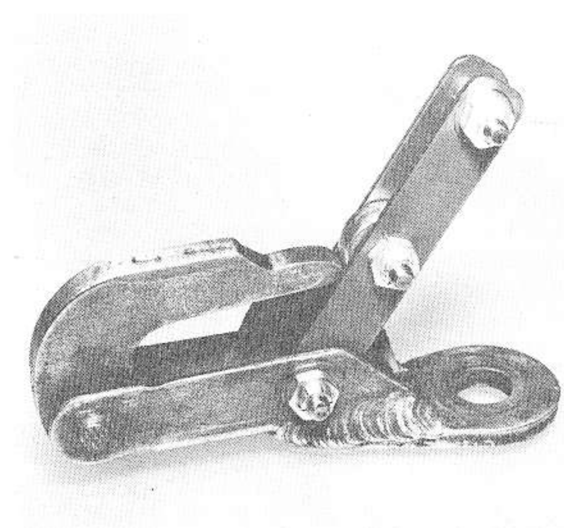
CONFIGURATION A



CONFIGURATION B



CONFIGURATION C



CONFIGURATION D

FIGURE 2a