OLEO UNDERCARRIAGE DESIGN

Abstract of Paper read by Mr. G. H. Dowty, Associate Member, at a meeting of the Institution held at the Engineers' Club, W.1., on November 17th, 1922. Mr. H. B. Molesworth in the Chair.

Mr. G. H. Dowty said:

An oleo landing gear generally consists of two or more telescopic legs, which are connected at their respective ends to the underside of the fuselage, or to the lower plane, and to the axle near the wheel hub. In the telescopic leg is fitted a piston, which, on landing, moves relatively to the wheels against a column of oil with a high pressure, and so doing definitely absorbs the energy.

It is well known that hydraulic resistances absorb an amount of energy which varies as the square of the speed. A hydraulic machine therefore may be employed as a brake, and it is in this way that large amounts of surplus energy are most easily disposed of. The hydraulic mechanism once used to work heavy guns on board ship is a type of apparatus which takes advantage of this system, for it serves as a very effective brake, and by simple adjustment readily controls the speed of movement.

A hydraulic brake is constructed by interposing a mass of fluid between the elements of a pair so that any motion of the pair causes a breaking-up of the fluid with a corresponding resistance.

There is no doubt that the use of hydraulic buffers with success on guns did much to stimulate their use on aircraft undercarriages. But although their early adoption certainly had a tendency to reduce bouncing, yet the weight went up to a prohibitive figure, and possibly this was the reason that prevented the general adoption of this type of undercarriage.

There are two distinct types of oleo shock absorbers. The one relies on absorbing the energy by means of a piston having variable orifices to permit a flow between the under and upper sides of the piston, while the other has leak orifices and a spring loaded valve, which prevents the internal cylinder pressure from rising above a pre-determined value.

It is, of course, advantageous to have a uniform resistance during the whole travel, because the maximum pressure in the cylinder would be diminished, and less strain thrown on the gear.

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The stroke of the leg is, of course, limited on each undercarriage by the conditions relating to that particular machine, but the larger the travel of the leg the better will be the shock-absorbing qualities. In a gear having a positive extension the stroke is definitely limited, and on most machines is from 6 to 8 inches; but if the non-positive type is used extensions of 14 or 16 inches can quite easily be obtained. The method in this type is to allow the oleo piston to do work against the column of oil for 7 or 8 inches before coming on to the rubbers. If the rubbers come into operation immediately on landing it is found that, although an oleo leg is incorporated, yet bouncing does still occur; but if the oleo leg has absorbed all, or even say 50 per cent., of the Ke before the rubbers come into play, then the oscillations or rebounding on landing can be greatly reduced.

To one accustomed to the two types of oleo there can be no doubt that the non-positive type possesses considerable advantages, and, in my opinion, this gear is without doubt the undercarriage with the best future before it.

In this type of leg the inner telescopic member carries the piston head. During the first part of the stroke the oil dashpot only is in operation, and this is supplemented by the rubber suspension rings during the final movement of the leg.

The oleo leg is never fully extended until after the machine is taken off, when the weight of the wheels and axle brings the legs down to landing position. The greater portion of the energy represented by the vertical velocity in landing is immediately disposed of by the oil dashpot before coming on to the rubber springing, thus avoiding bouncing or rebounding of the machine when alighting.

The objection to the non-positive return type is the possibility that the legs may stick and not return fully, but this is more apparent than real, for with experience of many such gears I can say that no such trouble has yet arisen. Care should be taken in design to make provision for an ample supply of oil to all sliding parts.

In first considerations for the lay-out of an oleo leg it is very desirable to have an approximate idea of the size of piston, area of ports, etc.

For this purpose graphs have been compiled from several well-known undercarriages.

The horizontal ordinate gives static load per leg in pounds, and the vertical ordinate piston areas in square inches. Superimposed is another graph with the same horizontal values, but with vertical ordinates giving effective area of leak holes.

From these graphs we may take the values given as a good average, and commence working on that basis.

The most important part in the design of any oleo gear is the arranging of the ratio of the area of the piston to the area of the leak orifices. This is a very delicate piece of work, as a difference of a couple of hundredths of an inch in the size of the orifice makes all the difference between a steady smooth action and a harsh one.

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The amount of resistance depends on the difference between the cross sectional area of the piston and leak ports, and by varying the ports at different points as required it is possible to adjust the resistance so as to be uniform throughout the whole stroke, thus keeping the maximum load on the fuselage structure as low as possible.

This work is difficult because very little is known theoretically about the behaviour of a fluid forced through a narrow orifice under sudden and violent pressure.

To find the amount of work any leg is capable of absorbing we require to know its construction, for there are several types, and each require a different style of investigation.

Generally all oleo legs can be divided into two classes, and each class can have two sub-divisions :—

(I.) Valveless type :
- (a) Leak orifices of constant area.
- (b) Leak orifices variable.

(II.) Relief valve type :
- (c) Valve limits maximum pressure.
- (d) Valve does not limit maximum pressure.

The latter case falls partially under the conditions of a, b, and c, but generally we can neglect it, as it is rarely met with, and I propose to show how the work done can be roughly calculated for any of the cases a, b, and c abovementioned.

For case (a) :—

This is the most simple construction of any of the legs, though not by any means the most effective.

If—

RH = Hydraulic resistance in pounds.
w = Weight of a cubic foot of oil.
A = Area of piston.
a = Effective area of leak orifices.
n = A/a.
V = Velocity of the piston in ft. per. sec.
g = 32.2.

Then at any time—

$$RH = w\ A\ (n-1)_2\ V^2/2g\ (1).$$

For the proof of this I must refer anyone to books dealing with hydraulics.

To represent this graphically—

Draw a curve on which the ordinate K N at any point N represents the retarding force (RH) at any time after the leg has contracted through the space O N from the point O at which the machine has its maximum vertical velocity. The area O A F D of this curve represents the kinetic energy which has all been absorbed by the hydraulic resistance of the leg.

Further, the area K N N′ K′ between two ordinates will represent the diminution of energy as the leg contracts through the space N N′ between them, a circumstance which enables us to construct the curb, for if V V′ be the vertical velocities of the machine at N N′ respectively

$$\text{Area K N N'K'} = W \quad (V^2 - V'^2)/2g. \quad (2).$$

But if RH,RH′ be the corresponding values of RH $KZ = RH - RH' = w\ A\ (n-1)^2\ V^2 - V^2/2g$. (3). And if the ordinates be taken near together the area in question will be nearly KN.NN′.

We have, therefore, by division—

$$\frac{KZ}{KN} = NN'\ w\ A\ (n-1)^2/W. \quad (4).$$

That is, if a number of equidistant ordinates be drawn near together the ratio of consecutive ordinates is constant. The curve may be roughly traced from this property.

Case (b) :—

Generally this type of leg takes the form of a taper needle, and we require so to arrange the taper that the resistance throughout the stroke is constant.

At any time the resistance is given as before by the following equations :—

$$RH = w\ A\ (n-1)^2\ V^2/2g\ (1);$$

but in order that the resistance may be constant we must have—

$$RH2 = w\ A\ (n-1)^2\ V^2/2g\ (5)$$

where RH2 = the mean resistance in lbs.,

so that $(n-1)\ V$ is constant.

Further, since the retardation is uniform,

$$V^2 = 2g. \frac{RH2}{W.}\ x \quad (6),$$

where x is the distance from the end of the stroke.

It appears, therefore, that the orifices should vary in such a way that $(n-1)^2x$ should be constant.

To illustrate this I have worked out the needle for a particular case, and the calculations are appended.

Case (c) :—

If a valve be fitted in the piston, so that whatever the vertical velocity, the load on the piston is kept below a certain value, then the maximum pressure on the piston head and cylinder walls will be known.

The valve will close immediately the pressure in the cylinder falls below a point when the valve spring overcomes the pressure on the valve.

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If we know at what part of the stroke this pressure occurs we can find the work done by the oleo during the time the valve is open.

Assuming that the valve opens at 1,300 lbs. per square inch, and that the piston area is A square inches—

Then $1{,}300\,A = K\,V^2$.

Taking $K = 360$.

Then velocity V, at which the valve will close, is given by

$V = \sqrt{1{,}300\,A / K} = 3.38$ feet, per sec.

When the valve is closed, and only the leak orifices open, the conditions are those of case (a). From this knowledge we are able to draw a composite graph showing the total work absorbed. This type of gear has advantages in that the maximum pressure on the fuselage structure is limited under ordinary conditions, but if the gear be called on to do extra work should the landing be extra severe, then it will fail; but with types of gear such as a and b the harder the landing the more work there will be absorbed. Against this, of course, is the fact that the maximum pressure is not limited. Individual designers must choose for themselves which in their opinion is the lesser of the evils, for both types of gears have their advantages and disadvantages.

In all calculations I have purposely left out all considerations of work absorbed by the rubber springing. In my opinion all the kinetic energy should be absorbed by the oil dashpot before the rubbers come into play. This will prevent bouncing and give a steady landing. The rubbers are only called into work for providing a springing when taxi-ing.

Oil.—The oil generally used in oleo legs is colz oil. This oil is quick flowing and comparatively cheap. Where the legs are subject to low temperatures glycerine should be used. With good fitting telescopic legs and a well-designed gland the oil rarely needs supplementing.

The weight of one cubic foot of colz oil is 56.99 lbs.

Weight of one gallon is 9.16 lbs.

In this paper I have not attempted to deal with anything other than the design of the oleo leg itself.

Comparatively little work has been done on this subject, and designers have not much data to help them, but I trust that what information this paper contains may be found of use to those engaged on this type of undercarriage.

In conclusion, I wish to thank Mr. Sloper, of Messrs. the Palmer Tyre Company, for permission to use the tyre load deflection graph, and Mr. H. B. Molesworth for reading the proofs of this paper.

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APPENDIX.

Weight of machine = 5,000 lbs.
Vertical velocity = 10 ft. per sec.
Kinetic energy = $mu^2/2g$ = 93,710 in lbs.

Where : m = Weight of machine,
u = Vertical velocity,
g = 32.2.

Assuming that the particular wheel used is capable of absorbing 15 per cent. of the Ke,

Then Ke to be absorbed per leg = $79{,}200/_2$ in lbs.
Let the stroke in this case be 10 ins.
Then the mean Hydraulic Resistance (RH_2) = 3,960 lbs.
Internal Cylinder Pressure = 1,260 lbs. per sq. in.
Where

$$RH_2 = w\,A\,(n-1)^2\,V^2/2g,$$

and w = 57,
g = 32.2,
A = 3.14.

$w\,A/2g$ = a constant = .0193.

Where D = distance in feet of the needle section from the piston.
a = cross-section area of needle in sq. ins.,
T = Taper of needle,
d = diameter of needle in ins.

TABLE I.

RH_2	$\frac{W_2}{RH_2}\cdot\frac{1}{2g}$	V^2	$1+\frac{\sqrt{RH}}{\cdot 0193V^2}$	A/n	$\frac{W_2\,.\,V^2}{RH_2\,.\,2g}$	D	.1104−a	$r^2=a_1/K$	d	T
3960	.00833	100	46	.068	.833	0	.0424	.0135	.234	—
		81	51	.061	.675	.158	.0494	.0157	.252	.018
		64	58	.054	.533	.3	.0564	.018	.268	.016
CONSTANT.	CONSTANT	49	65	.048	.408	.425	.0624	.0199	.284	.016
		36	76	.041	.30	.533	.0694	.022	.298	.014
		25	92	.034	.208	.625	.0764	.0244	.312	.014
		16	114	.027	.133	.7	.0834	.0265	.326	.014
		9	146	.021	.075	.758	.0894	.0285	.338	.012
		4	228	.014	.033	.8	.0963	.037	.35	.012
		1	460	.007	.008	.825	.1033	.0328	.362	.012

Calculations based on a leak orifice of $\frac{3}{8}$ in. diameter, i.e., .1104 sq. ins. area.

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