OPTIMUM DESIGN OF CONTROL SYSTEM IN RECOIL MECHANISMS

by

J. P. SIRPAL

Institute of Armament Studies, Kirkee

ABSTRACT

The various parameters in the design of a control system in a conventional type of recoil mechanism have been optimised for minimum weight and the ideal form of velocity space variation during runout has been derived. A theoretical analysis for some of the equipments in service reveals that their actual values are very nearly the same as those given by theory except for the ratio l by d_i which are lower than those theoretically suggested indicating that wide margins of safety against buckling have been used in their design.

Introduction

In a conventional type of recoil system a part of the recoil energy is stored in the recuperator which is utilized to push the recoiling masses back to their firing position. The recuperators are either the spring type or the hydropneumatic type and in either case it could be assumed that the accelerating force provided by the recuperator decreases linearly as the runout proceeds.

In order to bring the recoiling mass to rest without impact, it is obvious that their motion should be arrested towards the later part of runout. The braking force is supplied by the control system which is in the form of a solid steel plunger with a slight taper entering a cylinder full of oil. The control system can, therefore, be designed to give any required variation of the braking force by a suitable design of the taper rod which governs the area for the escape of oil at any instant of runout. The maximum braking force that can be brought into effect at any instant is, however, limited from stability considerations and therefore, for a constant factor of stability throughout runout, the net braking force, *i.e.* the algebraic addition of the actual braking force and the accelerating force, must decrease linearly as does the stabilizing moment and consequently the stabilising force due to the motion of the recoiling masses as the runout proceeds.

Analysis

The variation of the recuperator, control system, stabilising and the net braking forces are shown in figure 1, in which L and l refer to the total distance of runout and the length of the control cylinder. (The frictional and buffer forces are neglected, generally being small in comparison to other forces).

Let the recuperator and the control system forces be represented by the equations $S=S_{max}$ $(1 - \alpha_s$) and $F = F_{min}$ $(1+\beta x)$ where S_{max}, F_{min}, α and β are known constatnts for any equipment and s and x are positive in the directions indicated in figure 1.

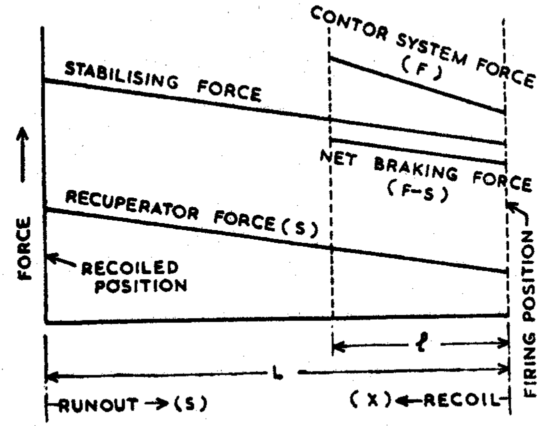


Fig. 1

The following equations can now be written based on various considerations that govern the design of a control system:—

(*a*) *Energy*: For the energy of the recoiling masses to be zero over the distance L, we have —

$$S_{max}\ L.\,(1-(\alpha/2)\,L) = F_{min}\ l\left(1+\frac{\beta}{2}\,l\right) \quad .. \quad .. \quad (1)$$

which determines the value of l. But since the values of α and β are of the same order and small in comparison to unity and $S_{max} >> F_{\min}$ and $l \gg L$, it is necessary that the recuperator force must be scaled down by a factor R by suitable means (by retard valve in hydropneumatic systems and buffer in case of spring recuperators). The above equation can, therefore, be rewritten as :

$$S_{max}\ L/R\left(1-\frac{\alpha}{2}\ L\right) = F_{min}\,l\left(1+\frac{\beta}{2}\,l\right) \quad .. \quad (1a)$$

(*b*) *Maximum Force* : The maximum steady force (F_{max}) that the control system exerts is equal to $F_{min}\,(1+\beta l)$. However, the system will have to be designed to withstand a force of twice this magnitude because of the reflected pulse from the closed end of the control cylinder when the control system suddenly comes into operation (Attenuation of the pressure wave within the medium is neglected). Therefore if d_i and d_e represent the internal and external diameters of the control cylinder, f_y the yield stress of the material, the maximum internal pressure that the cylinder can withstand without permanent diametral expansion is equal to $fy\,(k^2-1)/2k^2$ where $k = d_e^2/d_i$ (on the basis of maximum shear stress theory) and with a factor of safety equal to n the value of the maximum permissible internal pressure is given by the expression $p_{max} = (k^2-1)/2k^2.\ f_y/n$ and therefore from the consideration above, we have:

$$F_{max} = F_{min}\ \ (1+\beta l) = 1/2\left[\frac{\pi}{4}\,d_i^{\,2}\right] p_{max} \quad .. \quad (2)$$

(c) *Buckling*: The control plunger is subjected to an axial load which is maximum and equal to $2\,F_{max}$ at the instant the control system comes into operation which may cause buckling. The maximum load that a column such as a control plunger which is fixed at one end and guided in the control cylinder in the other can take is given by the Rankine Gordon Formula and is equal to $f_c\ A/[1 + 16a\ (l/di)^2]$ where f_c is the crushing strength of the material and can be taken to be equal to f_y, A the area of the X-section of the column and 'a' a constant which for such a column of steel is 1/15,000 and therefore if a factor of safety of n_1 is used the maximum axial pressure on the column must be limited to a value given by the equation:

$$p_{max} = \frac{f_y}{n_1\left[1 + \frac{16}{15{,}000}\left(\frac{l}{di}\right)^2\right]} \quad \ldots \quad (3)$$

(d) *Size*: The volume of the control system can be written as :

$$V = \frac{\pi}{4} d^2_e\, l = \frac{\pi}{4}\, k^2\, d^2_i\, l = 4n\, \frac{F_{min}\,(1 + \beta l)\cdot l}{f_y}\, \frac{k^4}{(k^2 - 1)}$$

with the help of equation (2). For a minimum volume $\frac{\partial v}{\partial k} = 0$ which gives $k = \sqrt{2}$ and defines the optimum X-section of the control cylinder. Hence the optimum value of p_{max} equals $f_y/4n$ and equations (2) and (3) can be rewritten as

$$F_{max} = F_{min}\ (1 + \beta l) = (\pi/32).\,(f_y/n).\, d_i^{\,2} \quad \ldots \quad (2a)$$

$$\text{and } l/d_i = 30{\cdot}5\sqrt{(4n/n_1) - 1} \quad \ldots \quad (3a)$$

The best choice of R, l, d_i and d_e for a minimum size of the control system can therefore be made with the help of the following equations :

$$(S_{max}\, L)/R\, (1 - \frac{a}{2} L) = F_{min}\, l\, (1 + \frac{\rho}{2} l)$$

$$F_{min}\ (1 + \beta l) = \frac{\pi}{32}\, \frac{f_y}{n} d_i^2$$

$$l/d_i = 30{\cdot}5\sqrt{(4n/n_1) - 1} \text{ and } d_e/d_i = \sqrt{2}.$$

The energy that the control system of optimised dimensions absorbs is given by:

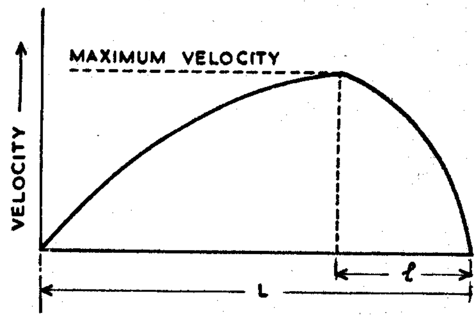
$$E = (S_{max}\, L)/R\, (1 - \frac{\alpha}{2} L)$$

and the maximum theoretical velocity with which the recoiling masses approach the control system is then given by :

$$v^2 = \frac{2g}{W}\, \frac{S_{max}\,(L - l)}{R}\, [1 - \frac{\alpha}{2}(L - l)]$$

where W is the weight of the recoiling masses.

The velocity space variation during run out is shown in fig. 2(a). In actual practice, however, it is found that in almost all equipments the recoiling masses attain the theoretical maximum velocity as calculated above by the end of that portion of runout which occurs under partial vacuum which is created in the buffer cylinder during recoil of the gun because of the inablili y of the replenisher oil to enter the buffer cylinder as fast as the buffer cylinder rod withdraws from the cylinder. It therefore follows that if a control system of optimised dimensions is to be used, the velocity of the recoiling mas es should not be allowed to exceed the theoretical maximum that the optimised control system can withstand and this leads to a stage of runout during which the velocity of runout is maintained constant as shown in fig. 2(b) by a suitable design of the retard valve.



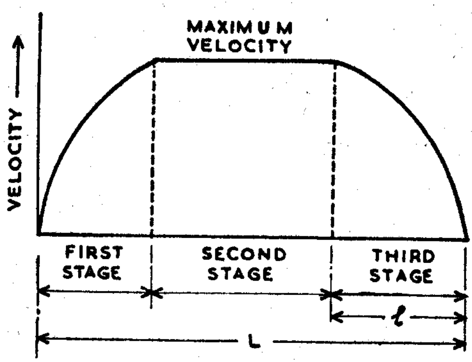


FIG. 2(a) and 2(b)

Table I gives a comparison of the actual and the theoretical values of the various control system parameters for some of the service equipments. The theoretical calculations were made by assuming $n = 1\cdot5$, $f_y = 17\cdot4$ *t.s.i.* and l/d_i equal to the actual values as used in the equipment under analysis. It may be seen that the actual values of l/d_i are quite low (of the order of 10) indicating thereby that the factor of safety against buckling of the plunger is quite high ($n_1 = 5\cdot5$ for $n = \angle\, 1\cdot5$). The actual values of l and d_i are slightly higher than those theoretically found which shows that lower pressures in the control system *i.e.* higher margins of safety than $1\cdot5$ have been used. The values of R in each equipment are indicative of the portion of the recuperator energy that is wasted out by the retard valve buffer and frictional forces.

TABLE I

Serial No.	Equipment		QF 25 pr How	QF 6 pr 7 cwt	QF 3·7″ How	BL 5·5″ Gun/How
1	S_{max}	lbs	3,675	1,200	2,475	12,675
2	S_{min}	,,	2,240	600	1,000	6,100
3	Length of recoil (L) ..	in	36	30	35	54
4	W	lbs	1,434	932	680	4,950
5	Reaction on spades when fully runout (R_1)	,,	125	177	150	385
6	Length wheel to spade (Z)	in	106·5	102	91	159
7	Trunnion height (h) ..	,,	45·5	30	33·5	52
8	Force at Trunnions required to overturn the eqpt about the point of wheel contact with ground when fully runout $=R_1\ Z/h$	lbs	290	600	407	1,177
9	Minimum control cylinder force for a factor of stability = 1.		290+	600+	407+	1,177+
	$F_{min} = R_1 Z/h + S_{min}/R$	lbs	2240/R	600/R	1000/R	6100/R
10	$\alpha = \dfrac{(S_{max} \quad S_{min})}{L.\,S_{max}}$	in^{-1}	0·011	0·017	0·017	0·010
11	$F_{min} \cdot \beta = (W/h + \alpha S_{max})$	lbs/in	72	51·5	62	222
12	l/d_i (Actual)		13·25	12·75	10·75	7·6
13	R		18·0	2·88*	14·5	24·4
14	l (*i*) Actual effective ..	in	10·6	10·2	6·45	9·50†
	(*ii*) Theo:	in	8·34	9·0	6·23	8·70
15	di (*i*) Actual	in	0·8	0·80	0·600	1·250
	(*ii*) Theo:	in	0·63	0·706	0·580	1·145
16	Maximum velocity of runout just before control system becomes operative— (*i*) Actual	ft/sec.	3·40	5·80	3·00	2·30
	(*ii*) Theo: ..	,,	4·23	6·43	5·32	4·44
17	Velocity of runout at the end of 1st stage.	,,	4·30	6·15	3·75	2·90
18	E_{max} (*i*) Actual.. ..	lbs	2300	1600	1450	4700
	(*ii*) Theo: ..	,,	1014	1272	1483	3357

*This gun employs spring recuperator.

† Higher grade of steel for the control system is used.

Acknowledgements

The author is thankful to Prof. C.H. Smith for the helpful suggestions given by him in the writing of this paper and to Mr. A. M. Subramaniam for the help he gave in the calculation work.

References

1. M.G.S. Publication *GM/9/45-Recoil*
2. Sirpal, J. P. *IAS research memorandum.* " *Analytical* .. .. ".