Table 2 Abstract of vibration formulae on design of the foundation

Vibration	Supporter		Kind of input force	
			Saw tooth shape	Rectangular shape
Anvil	Spring and damper		Formula (20)	Formula (7)
	Spring only		" (22)	" (8)
Foundation	Spring only			" (44)
	Conventional calculation	Spring & damper	Formula (55)	" (51)
		Spring only	" (56)	" (52)

By the formulae mentioned above, theoretical solution for the supporting method using the damper and spring is much complicated obviously. Though the formulae do not give an exactly absolute value for design of the foundation and mounting of Hammer-Forging Machine for industrial purpose, except strict consideration of the solution, under the assumption of being supported only by spring, it is very convenient to discuss with the formulae shown in Table 2 in accordance with each shape of input force.

The writer has acknowledged that the foundation

which has been designed by using the formulae of the conventional solution is very good and these formulae are accurate and can be applied for actual purpose.

In closing, the writer wishes to express his sincere thanks to Prof. emeritus H. Yamanouchi and Prof. T. Takahashi of Waseda University for their valuable advices in the investigation.

Reference

- (1) S.Omodaka : *Bulletin of JSME*, Vol.1, No.4(1958), p. 342 and p. 348.

621.733.544-24:531.39

Theoretical Analysis of the Tup-Motion of a Forging Air Hammer*

By Seihei SAKAI**

In this paper, as a whole the following five items with respect to air hammer could be discussed and their necessity is proclaimed.

- i. Theory and numerical calculations about characteristics of the tup-motion.
- ii. Adequate selection of the principal speed of a crank shaft.
- iii. Measurement for the forging effect of an air hammer by Heim method using a test piece of lead cylinder, or by the final velocity of a single blow of tup.
- iv. Comparative discussion about the experimental data described in reference (2) and the theoretical analysis of the tup-motion by our study.
- v. Computation of the efficiency of an air hammer and the motor horse power required to drive it.

1. Preface

Fundamental equation and its approximate solution about the characteristics of the tup-motion

described in the first report⁽¹⁾ can be summarized again as follows.

The tup-motion is a periodic motion related to the reciprocating motion of pump piston by mediation of compressed air in both cylinders, and it can be expressed by Newton's law of motion as follows.

* Received 3rd April, 1959.

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$$m \cdot \alpha = \pm P - m \cdot g$$

However, above equation is integral equation with functional P and it is difficult to obtain a general or particular solution. Consequently it has been tried to replace the equation with an algebraic simultaneous equation of higher order applying successive approximation of integral and their approximate solution was obtained.

At the same time, by numerical calculation, characteristics of the tup-motion, so called acceleration, velocity, and displacement of tup were calculated numerically and also indicator diagrams and indicated horse power were obtained.

Now in this case, several numerical calculations for a 500 kg air hammer of Bêché type which was manufactured by our design formerly as shown in Fig. 1, are attempted through the solution as follows.

Characteristics of the tup-motion are to be affected by the speed of a crank shaft, driving the pump piston.

And most adaptive or principal speed which means that one cycle of single blow accords with one cycle of pump piston, is to be found. Moreover, torque of crank shaft, air pressure and temperature in both cylinders are calculated.

And then referring to the desertation of experimental research "Untersuchung eines Schmiede-lufthammers" von Dr.-Ing. Fr. Keilholz, the writer has found that characteristic curves obtained by our analysis are similar to these experimental records obtained using reliable instruments with a pick up, an indicator, and a recorder.

2. Fundamental equation of the tup-motion

$$m \cdot \alpha = (A_b \cdot P_b - A_a \cdot P_a) - m \cdot g \quad \dots (1)$$

where

m : mass of tup

α : acceleration of tup

g : acceleration due to gravity

A_b and A_a : effective cross sectional area of lower and upper sides of tup cylinder.

P_b and P_a : air pressure in lower and upper sides of tup cylinder.

As denoted in the first report, parenthesized term in above equation expresses the force to raise the tup upwards, and it is in a state of equilibrium with the self weight and inertia force of the tup. However, according as compressed air in the tup and pump cylinders may be varied by polytropic change, the following equation is obtained.

$$P_b = (\kappa \cdot Q_b^{-\gamma} - 1) \text{ and } P_a = (\kappa' \cdot Q_a^{-\gamma} - 1)$$

Fig. 1 General view of a 500 kg Bêché type air hammer

A : Pump piston   B : Tup
C : Cushion chamber
U and D : Inlet of free air supplied

Fig. 2 Schema showing mechanisms of tup and pump piston

Consequently

$$P_b = \left[\kappa \left\{ \frac{A_b}{100^2} \cdot h + \frac{\xi \cdot A_b}{100^2} (H_0 - h_0) + C_b \right\}^{-\gamma} - 1 \right] \quad \dots (2)$$

$$P_a = \left[\kappa' \left\{ \frac{\psi \cdot \beta \cdot A_b}{100^2} \cdot h_0 + \frac{\psi \cdot A_b}{100^2} (H - h) + C_a \right\}^{-\gamma} - 1 \right] \quad \dots (3)$$

where

Q_a and Q_b : volume of upper and lower sides of tup and pump cylinders

γ : polytropic exponents for air

κ and κ' : air constants in lower and upper sides of tup and pump cylinders

h and h_0 : optional displacement of the tup and corresponding displacement of the pump piston

1 : 1 ata

In other words, κ and κ' are numerical values to

be determined under initial state in which the pump piston and tup begin to start downwards and upwards from upper and lower sides of each stroke respectively as shown in Fig. 2.

Now, the integral with respect to displacement of tup h is expressed by approximation as follows.

$$h_{r+1} = h_r + v_r \cdot \Delta t + (\alpha_{r+1} + \alpha_r) \frac{\Delta t^2}{4} \quad \dots\dots\dots (4)$$

where

v_r : velocity of tup with suffix r

Therefore, applying approximation of integral (4) to Eq. (1), it is converted to algebraic simultaneous equation of higher order as follows :

$$\alpha_{r+1} = \frac{A_b}{m} \left[\kappa \left\{ \frac{A_b}{100^2} \left(h_r + v_r \cdot \Delta t + (\alpha_{r+1} + \alpha_r) \frac{\Delta t^2}{4} \right) + \frac{\xi \cdot A_b}{100^2} (H_0 - h_0) + C_b \right\}^{-r} - 1 \right] \\ - \frac{\psi \cdot A_b}{m} \left[\kappa' \left\{ \frac{\psi \cdot A_b}{100^2} \left(H - (h_r + v_r \cdot \Delta t) - (\alpha_{r+1} + \alpha_r) \frac{\Delta t^2}{4} \right) + \frac{\beta \cdot \psi \cdot A_b}{100^2} h_0 + C_a \right\}^{-r} - 1 \right] - g \quad \dots\dots\dots (5)$$

where suffix r is limited within $0 \leq r \leq n-1$.

Also, notations in above equation are the same as those denoted in item 3.

The above Eq. (5) is a simultaneous equation containing unknown quantities $\alpha_1 \dots \alpha_{24}$ and is able to be solved successively by means of successive approximation.

3. Numerical examples for a 500 kg air hammer

1. tup-weight		625 kg	500 kg
2. A_a		1 738 cm ²	ditto
3. A_b		938 cm ²	$\frac{A_a}{A_b} = \psi$
4. max. stroke of tup		$h_{\max}$ mm	$h'_{\max}$ mm
5. effective cross sectional area of upper side of pump cylinder	$\beta \cdot A_a$	2 050 cm ²	〃
6. ditto of lower side of pump cylinder	$\xi \cdot A_b$	1 510 cm ²	〃
7. stroke of pump piston	H_0	400 mm	〃
8. ratio of crank arm and length of connecting rod	$\lambda = r/l$	1/7	〃
9. No. of r.p.m. of crank shaft		$N \begin{cases} 120 \\ 60 \\ 180 \end{cases}$	〃
10. motor horse power		50 HP	〃
		(induction motor)	
11. clearance volume of tup and pump cylinders, upper side	C_a	0.022 1 m ³	〃
12. ditto, lower side	C_b	0.015 m ³	〃
13. Air constant in lower side of both cylinders	κ	0.032	〃
14. ditto, in upper side of both cylinders	κ'	0.083	〃
15. Period per cycle of the tup-motion	$\frac{2\pi}{\omega}$ s/c		〃
where ω : angular velocity rad/sec			
16. Minute time whose period is divided by n	$\Delta t = \frac{60}{N \cdot n}$		〃
where $n : 24$			
17. Polytropic exponent	$\gamma = 1.35$		〃

Substituting values of several constants denoted above, Eq. (5) can be solved with regard to the following terms (A) and (B). And thereby Figs. 3 and 4 are obtained.

(A)·I In case of $N=120$ r.p.m. and tup-weight=500 kg

i) Displacement of pump piston h_0 at stand-up point of tup.

$$g = \frac{A_b}{m} \left[\kappa \left\{ \xi \cdot \frac{A_b}{100^2} (H_0 - h_0) + C_b \right\}^{-r} - 1 \right] - \frac{\psi \cdot A_b}{m} \left[\kappa' \left\{ \frac{\psi \cdot A_b}{100^2} H + \frac{\beta \cdot \psi \cdot A_b}{100^2} h_0 + C_a \right\}^{-r} - 1 \right]$$

where $m = \frac{500}{9.8} \text{ kg} \cdot \text{s}^2/\text{m}$

Solution : $h_0 = 110.7$ mm and θ_4 corresponding to $h_0 = 60^\circ$

ii) α_5, v_5 and h_5 of tup at crank angle θ_5

$$\alpha_5 = \frac{A_b}{m} \left[\kappa \left\{ \frac{A_b}{100^2} \frac{\Delta t^2}{4} \alpha_5 + \frac{A_b}{100^2} (H_0 - h_{05}) + C_b \right\}^{-\tau} - 1 \right] - \frac{\psi \cdot A_b}{m} \left[\kappa' \left\{ \frac{\psi \cdot A_b}{100^2} \left(H - \frac{\Delta t^2}{4} \alpha_5 \right) + \frac{\beta \cdot \psi \cdot A_b}{100^2} h_{05} + C_a \right\}^{-\tau} - 1 \right] - g$$

$$\text{where } \Delta t = \frac{1}{48} \text{ sec}$$

$$\text{Solution: } \alpha_5 = 7.4 \text{ m/sec}^2, v_5 = \frac{\alpha_5}{2} \cdot \Delta t = 0.077 \text{ m/sec and } h_5 = \frac{\Delta t^2}{4} \cdot \alpha_5 = 0.8 \text{ mm}$$

And then calculations are repeated one by one from crank angle θ_6 to θ_{24} and the values for α , v and h corresponding to respective crank angle are obtained successively.

(A)·II In case of $N=120$ r.p.m. and tup-weight=625 kg

These terms were denoted already in the first report.

(B). In case of N r.p.m.=60, 120, and 180, and tup-weight=625 kg.

Characteristic curves h'' , h , and h''' in Fig. 4 correspond to the cases of $N=60, 120$, and 180 respectively, and each period per blow of tup does respectively to crank angle $\theta^\circ=294, 370$ and 450 .

Of course, crank angles advanced per blow for $N=60$ are too short, conversely for $N=180$ are too long.

Fig. 3 Characteristic curves: α - θ , v - θ , h - θ

Fig. 4 Characteristic curves under various speeds of crank shaft: α - θ , v - θ , h - θ

In consequence, the most suitable period of a blow should coincide periodically with the time at which the pump piston arrives at the upper dead point of crank angle. Therefore, $N=120$ is nearly equal to the principal speed. Also, the final velocity v m/sec of tup corresponding to $N=120$ is maximum above all. Namely $v=8.313$, $v'=4.69$, and $v''=7.235$.

As shown in Fig. 4, in case of $N=180$, it is obvious that the h - θ curve described by the first blow of tup is not identical perfectly with one which is described by the succeeding blow. Likewise, in the diagram S_B - α of Fig. 5, two kinds of "d" curves denoted with solid and dotted lines exist. However, the period of a blow expressed by the "c" curve in the same figure is just equivalent to crank angle $\theta=360^\circ$, and its corresponding speed is equal to $N=200$, so called "Haupt-Betriebs Drehzahl" ⁽²⁾.

Next, comparison of characteristics for two kinds of tup-weight is made as follows.

tup-weight:

$m \cdot g = 500 \text{ kg and } 625 \text{ kg}$

Crank angle at stand-up point of tup:

$$\theta = 60^\circ \quad 65^\circ$$

Max. acceleration of tup :

$$\alpha' = 130 \text{ m/sec}^2 \quad \alpha = 109.5 \text{ m/sec}^2$$

Max. velocity of tup :

$$v' = 0.78 \text{ m/sec} \quad v = 8.313 \text{ m/sec}$$

Max. stroke of tup :

$$h' = 719.3 \text{ mm} \quad h = 679.19 \text{ mm}$$

Crank angle corresponding to period of single blow of tup :

Fig. 5 Experimental records of $S_B-\alpha$ curves under various speeds of crank shaft

Fig. 6 Indicator diagrams in tup cylinder under adaptive r.p.m.

4. Air pressure indicated in tup and pump cylinders

Air pressure is calculated by the following expression and as the result, Figs. 6 and 7 are obtained.

Air pressure in upper side of tup cylinder.

$$P_a \text{ kg/cm}^2 = \frac{\kappa'}{[C_a + A_a(H-h) + \beta \cdot A_a \cdot h_0]^{\gamma}} - 1 \quad \dots\dots\dots(6)$$

Ditto in lower side of tup cylinder

$$P_b \text{ kg/cm}^2 = \frac{\kappa}{[C_b + \xi \cdot A_b(H_0-h_0) + A_b \cdot h]^{\gamma}} - 1 \quad \dots\dots\dots(7)$$

Fig. 7 Indicator diagrams in pump cylinder under adaptive r.p.m.

Table 1

tup-weight 625 kg	$N=60 \text{ r.p.m.}$	$N=180 \text{ r.p.m.}$	remarks
Crank angle mark	8 $\left\{ \begin{array}{l} \alpha'' = 18.1 \text{ m/sec}^2 \\ P_b = 1.5 \text{ kg/cm}^2 \text{ (gauge)} \end{array} \right.$	Crank angle mark 11 $\left\{ \begin{array}{l} \alpha''' = 52.1 \text{ m/sec}^2 \\ P_b = 3.62 \text{ kg/cm}^2 \text{ (gauge)} \end{array} \right.$	tup raised upwards
" mark	15 $\left\{ \begin{array}{l} \alpha'' = 32.27 \text{ m/sec}^2 \\ P_a = 0.75 \text{ kg/cm}^2 \text{ (gauge)} \end{array} \right.$	" mark 23 $\left\{ \begin{array}{l} \alpha''' = 131.8 \text{ m/sec}^2 \\ P_a = 4.2 \text{ kg/cm}^2 \text{ (gauge)} \end{array} \right.$	tup blown downwards
" mark	14 $\left\{ \begin{array}{l} h'' = 646.4 \text{ mm} \\ h_0 = 376.76 \text{ mm} \end{array} \right.$	" mark 22 $\left\{ \begin{array}{l} h''' = 579.4 \text{ mm} \\ h_0 = 30.36 \text{ mm} \end{array} \right.$	

Above diagrams denote one in which $N=120$ and tup-weight is 500 kg and 625 kg respectively. And then, Fig. 8 gives experimental data where $N=172$ and tup-weight is nearly 50 kg. Referring to these indicator diagrams, we can understand that those have a common feature in theory and practice. On the other hand, an extraordinary change of indicating pressure of air occurred due to an air cushion at the head of tup cylinder, is recorded actually in the diagram P_3-S_B of Fig. 8.

Next, mean effective pressure P_m and indicated horse power are obtained. By comparing indicator diagrams of Figs. 9 and 10 with acceleration and displacement curves of Fig. 4, the following results could be gained as shown in Table 1.

5. Crank torque M_d

Crank torque is calculated by the following expression and thereby Fig. 11 is obtained:

$$M_d = P \cdot r \frac{\sin(\theta + \varphi)}{\cos \varphi} \quad (8)$$

where

θ : crank angle

P : air pressure acting on pump piston

φ : obliquity of connecting rod

r : crank arm

Comparison of "c" curve marked (○) in which weight and inertia force of moving parts of pump piston are considered, with "a" curve in which

Fig. 8 Experimental records of $S_B-\alpha$, $M_d-\alpha$, $P_1-\alpha$ diagrams

Fig. 9 Indicator diagram in tup cylinder under two kinds of r.p.m.

Fig. 10 Indicator diagrams in pump cylinder under two kinds of r.p.m.

they are not considered, is clear as shown in the figure. Also, negative torque at crank angle $\theta = 195^\circ$ in the same figure is dependent upon the difference of air pressure $P_a = -0.076 \text{ kg/cm}^2$ and $P_b = 0.93 \text{ kg/cm}^2$ at crank angle θ_{13} in the indicator diagram of Fig. 7.

The diagram $M_a-\alpha$ of Fig. 8 is the record of experimental data and it is quite similar to one in Fig. 11 obtained by our study.

6. Air temperature in tup and pump cylinders

Air temperature is calculated by the following expression and thereby Fig. 13 is obtained.

$$T_1/T_2 = (P_1/P_2)^{(T-1)/T} \quad (9)$$

where

T_2 : absolute temperature of free air

P_2 : absolute pressure of free air

T_1 : absolute air temperature at optional point of crank shaft

P_1 : absolute air pressure at ditto

Fig. 13 indicates $t_a-\theta$ and $t_b-\theta$ diagrams which are calculated under condition that tup is 500 kg and 625 kg in weight, and N is equal to 120

r.p.m. and temperature of suction air is assumed as 40°C . Moreover, Fig. 12 indicates $P_a-\theta$ and $P_b-\theta$ diagrams under the same condition.

Therefore, we can understand that Fig. 12 denoting the air pressure and Fig. 13 denoting the air temperature are similarly formed figures, and there is no time lag between them against the same crank angle. However, d and e diagrams of Fig. 14 indicate the air temperature in upper and lower part of tup cylinder, and solid lines among them denote the temperature measured by thermocouple, and dotted lines denote the temperature

Fig. 11 Crank-torque diagrams $M_a-\theta$

$N=120 \text{ r.p.m.}$
 $a : mg=625 \text{ kg}$
 $b : mg=500 \text{ kg}$
 Fig. 12 Indicator diagrams $P_a-\theta$

$N=120 \text{ r.p.m.}$
 $a : mg=625 \text{ kg}$
 $b : mg=500 \text{ kg}$
 Fig. 13 Air temperature diagrams $t_a-\theta$

Fig. 14 Experimental records of $S_B-\alpha$, $P_3-\alpha$, $t_3-\alpha$ diagrams

amended without time lag. Hence, the latter are similar to indicator diagrams b and c of Fig. 14. And the former with time lag due to heat transfer are actual records. As denoted in reference (2), thermal diffusivity from air in tup cylinder to thermocouple is explained by next expression.

$$\theta = \vartheta + C \cdot \frac{d\vartheta}{dt} \quad \text{and} \quad C = \frac{G \cdot c}{F \cdot a}$$

where

θ : temperature of air in tup cylinder at measuring point

$\vartheta \cdot \frac{d\vartheta}{dt}$: temperature of thermocouple and temperature gradient with respect to time

G : weight of elements of thermocouple

c : specific heat of ditto

F : surface area of heat transfer

a : coefficient of heat transfer from air in tup cylinder to elements of thermocouple

This experiment should be done after the temperature of cylinder wall became steady in the range $60^\circ \sim 65^\circ \text{C}$, and diameter of copper and constantan element for thermocouple is 0.04 mm.

B : Recording wax paper for indicator diagrams

O : Oscillograph with recording camera

M_d : Coupling for measuring crank-torque

S_A : Recording pen for measuring piston-stroke

S_B : Sliding contact piece and electric wire for measuring tup-stroke

$P_1 \dots P_4$: Indicators with recording pen

$\theta_1 \dots \theta_4$: Thermocouples

Fig. 15 Experimental apparatus for air hammer

7. Measuring performance-instruments and efficiency of air hammer denoted in reference (2)

As shown in schema Fig. 15.

i) Air pressure in tup cylinder is measured by wire strain gauge and is represented on pen writing oscillograph in the form of waves. It is composed of following elements :

- Pick-up of air pressure or pressure head which consists of a wire strain gauge, a Wheatstone bridge and a power source.
- Galvanometer operating the writing pen.
- Recorder with wax paper for pressure curves.

ii) Stroke of pump piston is recorded by means of pen writing oscillograph by the same method as i).

iii) Torque of crank shaft is converted by the wire strain gauge and is represented on the fluorescent screen of Braun tube oscillograph in the form of waves, and also recorded by a camera. It is composed of the elements as shown in the connection diagram Fig. 16.

W : Crank shaft

D_1 : Active wire strain gauges measuring torque

D_2 : Dummy gauges compensating temperature

S : Slip rings

S_b : Brush for slip rings

B : Measuring apparatus of dynamic strain

G : 5000 c/sec transmitter

V : Amplifier

P : Phase discriminant

S_i : Filter

K : Cathode ray oscillograph

O : Oscillograph with recording camera

Fig. 16 Connection diagram of experimental apparatus measuring crank-torque

U : Electric source

D : Electric wire for sliding contact piece

K : Sliding contact piece attached to tup

O : Oscillograph with recording camera

S_B : Stroke of tup

Fig. 17 Connection diagram of experimental apparatus measuring stroke of tup

iv) Air temperature in the tup cylinder is measured by thermocouples and is represented on the recording camera.

v) Stroke of tup is measured by change of voltage and is represented on the recording camera. It is composed of the elements as shown in the connection diagram Fig. 17.

Efficiency of air hammer in reference (2) is shown nearly as follows.

	N r.p.m. = 200	175
E_{Md} : energy exerted by torque of crank shaft	234.5 kg m	170 kg m
E_{Sch} : energy exerted by single blow of tup	115.5 kg m	75 kg m
η_{Luft} : efficiency due to thermo-dynamic loss and leakage loss of air	60%	53%
$\eta_{Bär}$: efficiency due to frictional loss for moving the tup	83%	83%
η_{Sch} : efficiency due to energy loss for re-bounds of tup and for dies		

8. Computation of hypothetic efficiency of a 500 kg air hammer and of horse power required to drive it

In case of $m \cdot g = 625$ kg, $N = 120$ r.p.m., corresponding to item (A) · II

E_{Ni} : theoretical energy converted I.H.P. per rev. into energy per blow of tup

$$= \frac{N_i \cdot 4500}{N} \div 2360 \text{ kg m}$$

E_v : theoretical energy exerted by final velocity of single blow of tup = $\frac{1}{2} m \cdot v^2 \div 2350 \text{ kg m}$

A_{Heim} : actual energy exerted by single blow of tup, using Heim method assured by Bêché G.m.b.H. = $d^3 \{2.7a + 4.0(a^2 + a^4)\} \div 1230 \text{ kg m}$ where

$$N_i = 63.2 \text{ IP}$$

$$\begin{cases} N_i(t \cdot a) = \frac{A_a \cdot P_{m(t \cdot a)} \cdot H \cdot N}{4500} = 41.5 \text{ IP} \\ N_i(t \cdot b) = \frac{A_b \cdot P_{m(t \cdot b)} \cdot H \cdot N}{4500} = 21.7 \text{ IP} \end{cases}$$

$$P_{m(t \cdot a)} = 1.325 \text{ kg/cm}^2,$$

$$P_{m(t \cdot b)} = 1.285 \text{ kg/cm}^2$$

v : final velocity of single blow = 8.6 m/sec

d : diameter of lead cylinder = 8.1 cm

h : original height of ditto = 12 cm

h_1 : final height of ditto = 6.4 cm

$$a : \text{forging effect} = \frac{h - h_1}{h} = 0.467$$

Therefore, hypothetic efficiency of air hammer

$$\eta = \frac{A_{Heim}}{E_{Ni}} = 52\% = \eta_{Luft} \cdot \eta_{Bär} \cdot \eta_{Sch}$$

where

$$\eta_{Luft} \div 64\%, \quad \eta_{Bär} \cdot \eta_{Sch} \div 81.5\%$$

$$N_e : \text{horse power required to blow} = \frac{N_i}{\eta_m} = 74.3 \text{ IP}$$

$$N_M : \text{rated horse power} = \sqrt{\frac{\sum (N^2 \cdot t)}{T}} \div 50 \text{ IP}$$

(continuous rating)

where

$$\eta_m : \text{mechanical efficiency} = 85\%$$

t : time during hammer operation

T : whole time during one duty cycle

N_M is to be determined by means of root mean square for equivalent quantities of heat generated under variation of load during one duty cycle.

9. Conclusion

The data of numerical calculation obtained lately through approximate solution for theoretical analysis of the tup-motion which the writer has published in 1948, are unexpectedly similar to the experimental data of study at Hannover Technical College which Dr.-Ing. Fr. Keilholz has published in 1955. Besides, as the point of issue, the most suitable speed of crank shaft for air hammer having certain capacity is ascertained to be obtainable theoretically as explained above.

Hitherto, it becomes popularly known that the max. forging effect of tup is measured by a full blow on the test piece of lead cylinder. However, in reference (2), the experimental data are obtained by means of precise measuring instruments. Recently degree of precision in the home-made instruments has been improved, therefore if we choose appropriately the method of pick up, experimental data will be obtainable equally by means of an electro-magnetic or a Braun tube oscillograph with a recording camera or a pen writing oscillograph.

In closing, the writer apologizes for having quoted necessary data and figures from reference (2) without permission. Besides, the writer wishes to express his gratitude to Kohji Kondo, chief, Planning Dept. of Fuji Motors Corporation for his kind advice.

References

- (1) S. Sakai : *Trans. Japan Soc. Mech. Engrs.*, Vol. 14, No. 47 (1948), p. 1-115.
- (2) Fr. Keilholz : *Werkstattstech. u. Maschinenb.*, Jg. 45, Ht. 4 (1955), S. 181.

Discussion

S. OMODAKA : (1) Is it not necessary to consider the condition of re-bounding in clause 3?

For, between the last and the beginning of hammer-tup motion, you assumed tup is stopped. This means tup has no displacement, velocity and acceleration at the beginning of motion.

(2) (a) What characteristic has the Electro-magnetic Oscillograph which was used for P_{3-4} - S_B curve in clause 4?

(b) You say, lead cylinder can be used for the measurement of hammering effect in clause 9.

In actual work, I doubt, the forging effect for steel can be deduced directly from the tested value for lead.

Author's closure

(1) We can't answer your question without experimental data of rebounding phenomena. Even if initial condition of the tup-motion may be varied more or less by rebounds of tup, characteristic curves of the tup-motion are not considered to be affected remarkably by such transitional phenomena. Also in experimental curves S_B - α of Figs. 5 and 8, above phenomena are not observed.

(2) (a) Socks indicator diagrams P_{3-4} - S_B in Fig. 8 are not recorded by indicators P_{3-4} with a recording pen, and diagrams in Fig. 14 are rather recorded with one. Perhaps, socks i.d. may be obtained by using simultaneously an oscillograph of duplex Braun tubes which are connected with output of each indicator which picked up both diagrams S_B - α and P_{3-4} - α in Fig. 14.

(b) Lead cylinder is used as usual for measuring the forging effect and the effect approved by Béch G.m.b.H. are as shown in Append-Table 2. We have no preparation for your question about this item. However, experimental data by S. Omodaka can be explained for the question as shown in Append-Table 1, which proves that the relation between steel and lead with respect to forging effect under the same condition may be in definite ratio.

Questions at the conference

S. OMODAKA : (1) Refer to question (1)

(2) You say, forging effect by Heim's method is very reliable in clause 8.

What is your opinion about the points of Heim's formula criticized by K. Lange, also about the Lange's formula?

(3) Refer to question (2) (b)

I have already received your answer, and again ask you the following. As to your answer for my question (1)—

On my experiment for lead and steel which is done by 1/2 and 1 ton steam hammers, and for steel done by a 3 ton one, re-bounding phenomena are clearly recorded Braun-oscillograph and also by high speed photograph (about 1 200 frames/sec). Comparing my experimental record of tup-motion in (time-input) with Fig. 4 of yours I think it is necessary to take the re-bounding phenomena into consideration for the theoretical analysis.

Your answer is : as no re-bounding phenomena being found in S_B - α diagram of Fig. 8 and Fig. 5, the characteristic curve is not so much effected.

But unless clarification of the following is made, it shall be unable to determine by only two diagrams that re-bounding can be theoretically negligible, that is :

(a) It is not accidental that re-bounding phenomena in the experiment of above mentioned record were not observed.

(b) This record can be representative of all the forging phenomena.

(c) The measurement apparatus has accuracy enough to check out re-bounding phenomena, etc.

Concerning question (2)

Asking about this at the conference, I remember being given no clear answer and no letter either. Please give me answer.

Concerning question (3)

My question was, how do you relate and solve the difference of dynamic deforming resistance in

Append-Table 1 (refer to *Bulletin of JSME*, Vol. 1, No. 4 (1958), p. 342.)

Material of test pieces	Temperature of ditto °C	Size of ditto $d \times h$ mm	Anvil weight ratio and tup-weight 75 kg	Falling height of tup mm	Forging effect %	Spring Constant K kg/mm
Steel (SS41) T.C 0.18 %	900	43×80	15	2 000	8.2	53 300
Pure lead (99.58 %Pb)	room temperature	"	"	"	34.5	"

Append-Table 2

Tup-weight kg	Kinetic energy of single blow E_0 kgm	Size of test pieces $d \times h$ mm	Volume of ditto V cm ³	Specific resistance of strain E_0/V kgm/cm ³	Forging effect %
40	53	30 × 44	31	1.71	38.6
500	1 230	81 × 120	620	1.97	46.7
3 000	7 000	152 × 229	4 150	1.68	44.7

lead and steel. I understand by your answer that you recognized my theory that there exists definite relation between steel and lead in forging under same condition to some extent, and so resistance can be compared with each other under these relations.

But, it is difficult to examine the forging effect of steel by using a lead cylinder unconditionally. The following conditions shall be clarified.

(I) It is impossible to experiment on steel and lead with same conditions to infinite extent. There exist some limit of input energy for forging lead.

The condition on how to treat the experiment over this limit for lead.

(II) Influence on the forging effect which is exerted by the difference of dimension and shape of test piece for steel and lead (Heim says $d/h = 1/1.5$ etc).

Author's closure

(1) It is evident that characteristic curve of the tup-motion in Fig. 4 has a large period and displacement caused by a periodic oscillatory force, and as you say, re-bounding phenomena of tup due to elasticity of forging materials i.e. elastic oscillation which has a very small period and amplitude occur directly after a blow of tup. However, the latter is damped and vanishes in course of time. Experimental data of the phenomena are already given by S. Omodaka and T. Ikeshima. And we understand that such phenomena

owing to your experiment have been clearly recorded by high speed photograph. But they are not enough to analyze theoretically.

We can dare to answer as follows. Unless the equation about re-bounding phenomena of tup is solved and numerical values of period, amplitude, re-bounding velocity and damping process are obtained, we can't introduce the phenomena into the equation (5) described in this paper. In case of measuring the forging effect, it is conventional that single blow of tup is to be applied and continuous blows are to be avoided. On the other hand, experimental data of re-bounding energy and velocity of tup using a lead cylinder were studied already by H. Houben and Dr. H. Yamanouchi. But the amount of re-bounding energy is negligibly too small to compare with kinetic energy exerted by a blow.

(2) As regards Lange's formula and the points of Heim's formula criticized by K. Lange, we can't answer you as we have no preparation for this subject.

(3) With respect to questions of (I) and (II), we have no investigation about plastic theory of forging materials and their experimental data, therefore we can't answer on the subject.

Regardless of the comment about Heim's formula, the forging effect for the test piece of lead cylinder observed by Béch G.m.b.H. is shown in Append-Table 2, for max. 3 000 kg and min. 40 kg of tup-weight.