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Tubular Diesel Hammers

Extracted from EROFEEV, L.V.,
SMORODINOV, M.I., FEDOROV, B.S.,
VYAZOVIKII, V.N., and VILLUMSEN, V.V.
(1985) *Machines and Equipment for the
Installation of Shallow and Deep Foundations.*
(In Russian) Second Edition, Mashinostrenie,
Moscow, pp. 81-86

Tubular Diesel Hammers

Parameters for Free Falling Diesel Hammers

The tubular diesel hammer works as a two-stroke diesel engine. Although its operating principle is the same its construction is substantially different from crank-connecting rod diesel engines in the following ways:

- 1) Existence of the anvil, which is used simultaneously for sealing the working cylinder, adapting the hammer to various types of piles, and transmission of the force generated by the combustion of the fuel at the bottom of the ram stroke.
- 2) Absence of atomized fuel injection, as the fuel is sprayed into the cylinder during compression.
- 3) Change in the significance of the cycle strokes; with a normal diesel engine the working stroke is the expansion stroke, with a diesel hammer it is the compression stroke.

There are some important changes in the method of gas exchange.

In two-cycle diesel engines exhaust and fresh fuel intake take place at the end of the expansion stroke and at the beginning of compression at the outlet valves in the absence of combustion. In diesel hammers the piston (ram) speed increases with increasing penetration of the pile into the ground or, in other words, with increasing pile resistance. Because of this the conditions of air intake and blowdown are always changing.

With respect to blowdown subject to combustion and preparation for the next cycle, in design it is necessary to carefully choose the exhaust volume in conjunction with the exhaust port geometry and size, as the latter has the main influence for successful blowdown and subsequent filling with fresh air. The quality of blowdown depends upon the starting period, during the first strokes, when fuel is not being burnt fully. When soil resistance is low at the start of driving a significant part of the fuel energy is not being used to drive piling and for this reason the ram does not achieve full stroke. As a result of this the piston speed is low and the quality of blowdown is likewise.

The best blowdown during operation of a tubular diesel hammer is achieved by making the ratio of blowdown volume to working volume (the volume when the ports have just been closed) is 2.8-3.0. This is expressed as

$$2.8 \leq \frac{V_{pr}}{V_h} \leq 3.0$$

where V_{pr} = Blowdown volume, m³
 V_h = Working volume, m³

By fulfilling this condition the tubular diesel hammer works well, with a high quality of blowdown, good low temperature operation, and possible pile sets of 200 mm per blow.

The main sizes of the hammer are calculated in a way that at the maximum energy output the piston speed does not exceed 6 m/sec and thus damage the pile top.

In the tubular diesel hammers the fuel spraying is made without an injector. The fuel combustion chamber depends upon how the pile is struck and the ram stroke. At the end of driving, when pile set is nearly zero, the mean effective pressure p_i increases and exceeds the calculated pressure. The ram stroke increases as well and results in the maximum striking energy.

The existence of the anvil, the absence of atomized injection and other characteristics of the tubular diesel hammers influence the fuel combustion and it is possible to divide the fuel combustion into four phases:

- 1)** Delay of spontaneous combustion (the fuel induction period);
- 2)** Nearly instantaneous combustion of the fuel, with a rapid rise in both temperature and pressure in the combustion chamber;
- 3)** Period of slow burning;
- 4)** Expansion of the gases after combustion.

The most characteristic property of the tubular diesel hammer in operation is in the fuel induction period. This is the period in which the capability of the hammer for low temperature and soft soil starting is established. To improve the starting capabilities of the tubular diesel hammer at temperatures of -30°C and pile sets of 100 mm, the induction period must be raised to 3-5 msec. This can be achieved with proper selection of the fuel pump cam and plunger parameters.

Since in the tubular diesel hammer fuel spraying is made when the piston strikes the fuel pump cam, for high quality starting at low temperatures it is necessary to choose properly the compression ratio to be 15 under unchanged conditions of fuel spraying and mix formation.

The mix formation quality is for diesel engines one of the factors that determines proper spontaneous ignition and combustion conditions. In the tubular diesel hammer mix formation and combustion is allowed a short (2.5-5 msec) time. For effective mix formation it is necessary to have a high quality fuel mix plume, i.e., atomization of the fuel into small drops. It is possible to achieve a uniform microstructure of fuel drops and mix with uniform allocation of fuel in the combustion chamber. To fulfill this condition it is necessary to have a long fuel plume, enough fuel speed out of the fuel pump, proper air circulation, etc.

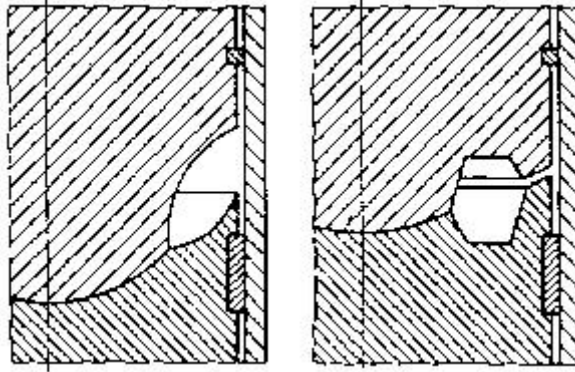


Figure 4.11

Drop shaped combustion chambers on tubular diesel hammers as shown on the left hand side in Figure 4.11 were made in the USSR at the beginning and now outside as well, but it does not satisfy the driving parameters. During impact fuel is lifted at high speed around the anvil and pressed by centrifugal force to its surface. This reduces uniform fuel allocation in the combustion chamber. Except for the fact that such a combustion chamber does not promote for intensive air circulation during compression, it decreases thermal efficiency during expansion.

Today tubular diesel hammers made in the USSR used a "gate" shaped combustion chamber which makes possible fuel lifting around the sphere after its ejection from the fuel pump, influenced by the diameter d . This is shown on the right hand side of Figure 4.11. This makes for intensive air circulation. In diesel hammers with similar combustion chambers air movement can be observed to be a mixture of three motions:

- radial (down from piston bottom) as a result of being pushed by the piston bottom during downward motion and compression;
- axial (out from cylinder centre axis) by both upward and downward motion of the piston.
- rotational motion in the combustion chamber as a result of air flow from under the piston to the combustion chamber.

Kinetic energy during air circulation seems to be an energy source for mix formation and is given by the expression

$$E = \frac{\rho_g F_K^3}{2f_K} \left(\frac{F_{por} - F_K}{F_{por}} \right)^3 \int_S^0 v^2 ds$$

where ρ_g = density of gases, kg/m³

F_K = space of spot of piston and anvil contact, m

f_K = projection of cylinder side surface with foundation, which equals to the diameter's spot of contact and height and/or equals depth of chamber, m

F_{por} = piston diameter, m

$(F_{por} - F_K) ds$ = volume of gas excluded from under piston during the time dt , m²

v = piston speed, m/sec

This expression shows that the smaller f_K is, i.e., the projection of the cylinder side surface (combustion chamber side surface), the higher the kinetic energy of flowing gases, and thus the mix formation of the gases is improved with decreasing ratio d_k/D_{ts} . For diesel hammers,

$$0.3 \leq \frac{d_k}{D_{ts}} \leq 0.35$$

where d_k = diameter of piston-anvil spot contact, m
 D_{ts} = cylinder diameter, m

Using this expression gives us the possibility of choosing basic parameters for combustion chamber size while aiming to get maximum energy for mix formation (see Table 4.16)

Table 4.16 Technical characteristics of symmetric “gate” shaped combustion unit for diesel hammers with various piston masses.

Parameters	Mass of striking parts, kg.					
	1250	1800	2500	3500	5000	7500
Cylinder Diameter, mm	300	345	400	470	550	660
Spot Contact Diameter, mm	102	120	155	170	200	270
External Gate Diameter, mm	240	155	325	370	450	540
Gate Radius, mm	35	35	46	56	60	61

Using these results we can also define the basic parameters of the tubular diesel hammer as follows.

Knowing the mass m_{sv} and space F_{sv} of the pile's cross section and using a maximum striking velocity of 6 m/sec, the maximum stroke is given as

$$H = \frac{v^2}{2g}$$

where g = acceleration due to gravity

The mass of the piston-ram is given from the expression

$$m_{u,ch} = \frac{m_{sv}}{2}$$

where $m_{u,ch}$ = piston-ram mass, kg
 m_{sv} = pile mass, kg

One cycle time of the hammer is given by the expression

$$t = \frac{4H}{g}$$

where t = total cycle time for the hammer, sec.

Of course the blows per minute for the hammer are

$$BPM = \frac{60}{t}$$

The mean effective pressure is given by

$$p_i = \frac{\eta_i \eta_v H_n p_o}{1.985 \alpha T_0 L'_0}$$

where p_i = mean effective pressure, MPa

α = surplus coefficient = 1.6

η_i = indicative surface coefficient = 0.04-0.45

η_v = coefficient of feeding = 0.7-0.8

H_n = specific fuel heat, kJ/kg = 43,260 kJ/kg

p_o = pressure at the beginning of combustion, MPa

for gate shaped combustion chambers = 0.11-0.12 MPa

for whirlwind type combustion chambers = 0.104-0.11 MPa

T_0 = absolute temperature of circulating air, °K = 293°K

L'_0 = theoretical air consumption for combustion of 1 kg of fuel, moles

This last quantity is given by the equation

$$L'_0 = \frac{1}{0.23} \frac{\frac{8}{3} g_C + 8g_{H_2} - g_{O_2}}{28.95}$$

where g_C = mass ratio of carbon content in 1 kg of fuel = 0.86 kg/kg fuel

g_{H_2} = mass ratio of hydrogen content in 1 kg of fuel = 0.13 kg/kg fuel

g_{O_2} = mass ratio of oxygen content in 1 kg of fuel = 0.01 kg/kg fuel

The mean effective pressure is also given by

$$p_{sr} = \eta_{mekh} p_i$$

where η_{mekh} = mechanical surface coefficient = 0.8-0.9

The striking energy of the piston-ram is

$$E_{ud} = \frac{m_{u,ch} v^2}{2}$$

where E_{ud} = striking energy of piston, J

and the working volume of the diesel hammer is

$$V_h = \frac{E_{ud}}{P_{sr}}$$

where V_h = working volume of diesel hammer, cm³

The cylinder diameter can then be computed by the equation

$$D_{ts} = \sqrt[3]{\frac{4V_h}{k\pi}}$$

where $k = s/D_{ts} = 0.95-0.65$ (for light diesel hammers k is higher and for heavier ones it is lower)

s = piston's working distance from time when exhaust port(s) are covered to impact, m

The combustion unit volume is computed by the equation

$$V_c = \frac{V_h}{\epsilon - 1}$$

where V_c = combustion chamber volume, m³

ϵ = compression ratio = 15

To test the starting quality of the tubular diesel hammer the coefficient of soil resistance is used as follows:

$$c = \frac{F_{ab} m_{sv}}{f_{ts} m_{u,ch}}$$

where c = starting quality = 2.5-3

F_{ab} = pile cross-sectional area, m²

$$f_{ts} = \frac{\pi D_{ts}}{4}, \text{ m}^2$$

Decreasing "c" lowers the starting quality while increasing "c" increases the length of the hammer.

The parameters for the fuel pump are controlled by the piling. Using the fuel-air relationships we have, the following

$$2.8 \leq \frac{\pi D_{is}^2 H}{4V_h} \leq 3.0$$

as we have seen before.

Parameter Selection for High Speed Diesel Hammers

Blows per minute from a diesel hammer increases by decreasing the stroke while maintaining the impact speed of 6 m/sec. For this reason, the high speed diesel hammer construction has a pneumatic bumper. The basic parameters of these high speed hammers, such as impact speed, striking energy and cycle time are given by the methods of G. I. Cherviakov and Yu. V. Beliaev, as parameters of lifting are calculated with respect to the above given method.

In a diesel hammer with a free falling piston the following parameters are known:

- V_c = combustion chamber volume
- F = area of piston
- s = piston's working distance
- $G_{d, ch}$ = mass of piston
- p_z = maximum pressure in combustion chamber
- n_1 = air compression constant
- n_2 = exhaust gas constant
- p_0 = atmospheric pressure
- \mathcal{E} = compression ratio
- s_{max} = maximum possible stroke of piston
- $l_m = V_c / F$ = combustion length of piston movement
- $l_\sigma = V_\sigma / F_\sigma$ = bumper chamber length
- V_σ = bumper chamber volume
- F_σ = bumper chamber area

Assuming the bumper chamber area and length, the bumper area is given as the difference between the area of the piston in the bumper and the area of the piston in the combustion chamber. Movement of the bumper is set from the hammer construction solution with respect to its maximum possible height.

The maximum air pressure in the bumper is

$$p_\sigma = p_0 \left(\left(\frac{1}{1-k_1} \right)^{n_1} - 1 \right)$$

where $k_1 = s_{\max}/l_\sigma$

The striking energy is given by the equation

$$E_{ud} = \bar{E}_{ud} G_{u, ch} l_\sigma$$

where

$$\bar{E}_{ud} = \frac{F_\sigma p_0}{G_{u, ch}} \left(\frac{1}{\varepsilon - 1} - \frac{F}{F_\sigma} \left(\frac{p_r}{p_0} + 1 \right) \frac{1}{n_2 - 1} \left(1 - \frac{1}{\varepsilon^{n_1 - 1}} \right) - \frac{\varepsilon}{n_1 - 1} (\varepsilon^{n_1 - 1} - 1) \right) - \frac{2G_{u, ch} k_1}{5F_\sigma p_0}$$

The impact speed is

$$v = \sqrt{\frac{2gE_{ud}}{G_{u, ch}}}$$

and the cycle time is

$$t = \frac{2s_{\max}}{v_{v_1}} + \frac{2(s_{\max} - s_v)}{v_{v_2}} + \frac{2s_v}{v_{v_2} + v}$$

where

$$v_{v_1} = \sqrt{2gl_\sigma \left(\frac{Fp_0}{G_{ud, ch}} \left(\frac{p_z}{p_0} + 1 \right) k_3 \frac{1}{\varepsilon - 1} \frac{1}{n_2 - 1} \left(1 - \frac{1}{\varepsilon^{n_2 - 1}} \right) \right. \\ \left. \left(1.2 + \frac{Fp_0}{G_{ud, ch}} \left(1 - \frac{F_\sigma}{F} \right) k_3 + \frac{Fp_0}{G_{ud, ch}} \frac{1}{n_1 - 1} \left(1 - \left(\frac{1}{1-k_3} \right)^{n_1 - 1} \right) \right) \right)}$$

and

$$v_{v_{21}} = \sqrt{2gl_\sigma \left(\frac{Fp_0}{G_{ud, ch}} \frac{1}{n-1} \left(\left(\frac{1}{1-k_1} \right)^{n_1 - 1} - \left(\frac{1}{1-k_3} \right)^{n_1 - 1} \right) + \right. \\ \left. \left(\frac{F_\sigma p_0}{G_{ud, ch}} - 0.8 \right) \frac{s_v - s_{\max}}{l_\sigma} \right)}$$

Blows per minute can be computed from impact time as before.