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Detailed computations with values of s/h (the ratio of the stroke to the connecting rod) between 0 and 1 have shown that its effect is so small that it can be neglected readily, and calculations with $s/h = 0$ suffice.

Determination of Constant A. The constant A is determined from the average conditions, i.e., from the values at $n = 0$. The average volume flow per unit time out of the cylinder must equal the volume flowing out, F_0, under average pressure p_0 or, from Equation [1]

$$\eta V_s \frac{\omega}{2\pi} = K p_0{}^r$$

Combining with Equation [9]

$$A = \gamma r \eta \frac{\omega}{2\pi} \frac{V_s}{V_{G0}}$$

Flow From Double-Acting Pumps. The effect of connecting-rod length cannot be neglected in these cases, and calculations are presented for $s/h = 0$ to 0.4 and for a range of values of head to crank-end piston-area ratios.

Such calculations were made for single-cylinder, duplex (90-deg crank) and triplex (120-deg crank) double-acting pumps, and the results are incorporated in Figs. 1 and 5.

These computations are lengthy and time-consuming and are not reported here since they are relatively well known.

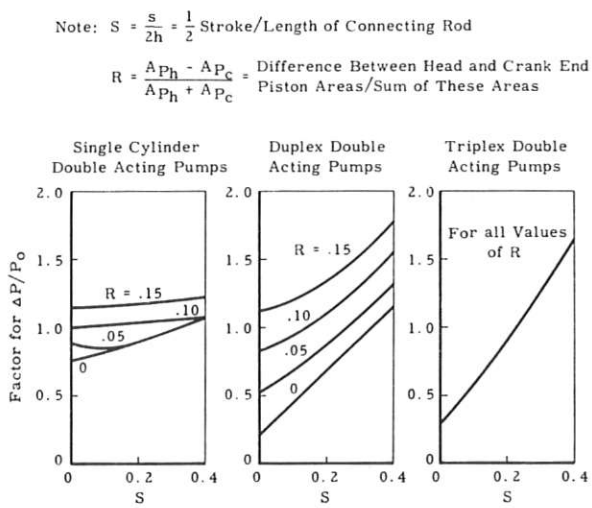

Fig. 5 Correction Factors for Pressure Amplitudes Computed From Fig. 1

Discussion

C. B. Haughton, Jr.[5] The gear-type pump is used extensively in the jet-engine fuel-control and accessory-testing field, where pump pulsations frequently affect not only the accessory being tested, but also pressure and flow-measurement apparatus in the rest of the system. Hence users of this equipment would be particularly interested in knowing whether or not the data of the authors' paper can be extended easily to the sizing of pulsation absorbers for gear-type pumps.

H. W. Iversen.[6] The design information presented for sizing gas-surge chambers covers the normal applications of reciprocating-pump types with air (or gases with $\gamma = 1.4$) in the chamber. The influence of the value of γ on the necessary gas volumes is of interest since gases other than air may be used in special cases, or the compression may deviate from the assumed isentropic conditions.

A simple analysis[7] is possible to show the influence of γ on the required gas volumes. Consider a single-cylinder single-acting pump with a surge tank, Fig. 6 of this discussion. The assumption of a small pressure variation in the surge tank (as in the paper) also specifies that the flow-rate variation is small in

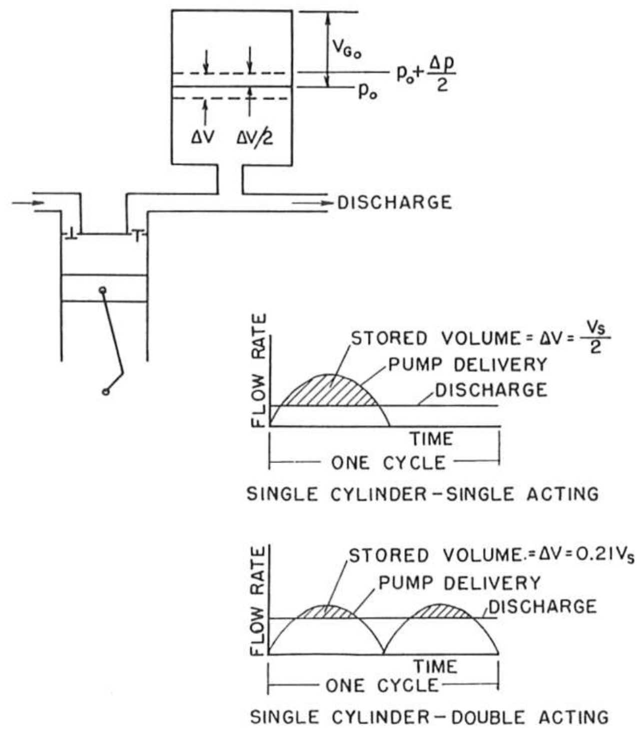

Fig. 6

the discharge line downstream of the surge tank. Since the pump delivers liquid for one half of the cycle, in order to have an essentially constant discharge-line flow rate, one half of the piston displacement must be stored in the surge tank to be released during the suction stroke of the pump. Thus the total change of gas volume in a cycle is one half of the piston displacement, and the change from the mean volume is one fourth the piston displacement. For the change of volume from the mean volume, the corresponding change of pressure is one half of the peak to peak pressure change.

Applying Equation [2] of the paper, for a polytropic compression

$$\left(p_0 + \frac{\Delta p}{2}\right)\left(V_{G0} - \frac{V_s}{4}\right)^n = p_0 V_{G0}{}^n \quad \text{.........[15]}$$

This reduces to

$$\frac{V_s}{V_{G0}} = 4\left[1 - \left(\frac{1}{1 + \frac{\Delta p}{2p_0}}\right)^{1/n}\right] \quad \text{..........[16]}$$

Table 2, herewith, shows the comparison between Equation

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[7] "Notes on the Theory of Pumping Machinery," by M. P. O'Brien and R. G. Folsom, University of California Press, Berkeley, Calif., Syllabus 1Q, 1934.

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[16] of this discussion and the authors' solution. Also shown in Table 2 are the gas volumes for isothermal compression with $n = 1$.

TABLE 2 VALUES OF V_s/V_{G0} FOR SELECTED PRESSURE PULSATIONS

$\frac{\Delta p}{p_0}$	V_s/V_{G0} Authors' Fig. 1 ($n = 1.4$)	Eq. [16] ($n = 1.4$)	Eq. [16] ($n = 1$)
	Single-cylinder—single-acting		
0.01	0.013	0.015	0.02
0.10	0.14	0.14	0.19
	Single-cylinder—double-acting		
0.01	0.065	0.07	0.095
0.10	0.6	0.66	0.91

The comparison of Equation [16] and the authors' results shows Equation [16] to be a reasonable approximation for the selected, single-cylinder, single-acting pump. For isothermal compression, considerably smaller gas volumes result. For other values of the polytropic exponent, Equation [16] may be used. It should be noted that the authors' results for isentropic compression are on the conservative side for actual systems where the compression is probably between isentropic and isothermal.

The simple analysis leading to Equation [16] also was applied to a single-cylinder double-acting pump with $S = 0$ and $R = 0$, Table 2. In this case the ratio of the liquid volume introduced to and released from the surge tank to the piston displacement is 0.21. The corresponding change in volume from the mean volume is one half of 0.21 or 0.105.

I. O. Miner.[8] The authors are to be complimented for presenting in such clear and concise a manner results which are believed accurate within the limits of their assumptions.

There is one common problem for which their assumptions do not apply. This is the case of metering the flow from reciprocating boiler feed pumps. Equation [1] of the paper can be modified to the following form for such installations

$$F_0 = K(p - p_b)^r$$

where p_b = boiler pressure.

If, as is generally the case, the boiler pressure is high relative to pipe friction, water inertia, and pressure drop through the boiler feed meter, a surge chamber sized by the authors' Fig. 1 might be almost useless.

If a restriction is placed between the surge chamber and the boiler, the effectiveness of the chamber can be restored to a considerable degree. Usually a valve is used, and it is closed slowly until surges are sufficiently reduced.

The authors also have neglected inertia. In some piping arrangements inertia tends to by-pass the surge chamber. It is preferable to mount the chamber on a tee, with the pump-discharge pipe pointing into the surge chamber. A baffle to prevent any water from entering the discharge pipe without first entering the surge chamber has been found to reduce the required size of the surge chamber.

The writer's company has long been recommending surge chambers with a gas volume of 5 to 6 times the displacement of one pump plunger except for single-cylinder, single-acting pumps. The greatest pulsations, in accordance with these recommendations, would be caused by a single-cyclinder, double-acting pump, and, according to Fig. 1, the pulsations would be 4 per cent. If we assume that the differential pressure across a flowmeter varies directly as the pressure from the surge chamber, the resulting error would be 1 per cent. If the piping is laid out as recommended, or more cylinders are used, the error will often be lower. Successful metering of liquids from reciprocating pumps for over 40 years indicates that the authors ' results can be relied on, provided the simple precautions set forth herein are taken.

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Authors' Closure

The authors wish to thank Messrs. Haughton, Iversen, and Miner for their interesting discussions.

In answer to Mr. Haughton's question regarding pulsation absorbers for gear pumps, the authors have not made a study of the flow-rate variation (F_i) from such a pump, but believe it would be fairly well approximated by a duplex double-acting pump with $R = 0$, $s/2h = 0$, and V_s = total discharge volume/number of teeth in one of the gears. This is true if the tip thickness of the gear teeth is small. Otherwise, some suitable value of $s/2h$ can be chosen to allow for the periods during which no discharge occurs from the gear.

Professor Iversen's comments are well taken and it is important to note that the use of a polytropic constant of 1.4 gives conservative results. The authors believe that, in general, the true behavior of the gas is more nearly isentropic than isothermal since the time available for heat transfer from and to the gas during a cycle is usually small.

Mr. Miner's first comment regarding the form of F_0 is most important and stresses a point not sufficiently well clarified in the paper, i.e., the neglect of the downstream pressure. However, the authors differ with Mr. Miner's conclusion. It should have been made clear that taking account of the average downstream pressure does not alter the effect of the surge chamber provided again that the remaining pulsation amplitude is small. This can be shown as follows:

Replacing Equation [1] by the form Mr. Miner suggests, we have $F_0 = K(p - p_b)^r = Kp^r (1 - p_b/p)^r = Kp^r [1 - (p_b/p_0)(1 + \Delta p/p_0 + \ldots)]^r$ which is very nearly $= Kp^r (1 - p_b/p_0)^r$ as long as pulsations are reduced to a value sufficiently small to make $\Delta p/p_0 \ll 1$. Then $F_0 = K^1 p^r$ where $K^1 = K(1 - p_b/p_0)^r$ is the constant that replaces K throughout the development without changing any of the conclusions.

Mr. Miner's suggestion regarding the design of the line-surge chamber combination is well taken and is a method of implementing No. 1 of the assumptions in the paper.

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