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Comparison of Various Dampers Effect on the Reciprocating Compressor Work the Pressure Pulsations Quantity in an Discharge Pipeline

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COMPARISON OF VARIOUS DAMPERS EFFECT ON THE RECIPROCATING COMPRESSOR WORK AND THE PRESSURE PULSATION QUANTITY IN AN DISCHARGE PIPELINE.

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ABSTRACT

The paper presents the pressure pulsation dampers of small hydraulic resistance and the most common criteria for the evaluation of their damping efficiency. The results of experiments on the effect of the location and the inside structure of dampers on damping efficiency in a discharge pipeline on the work of the reciprocating compressor have been presented. Some of these experiments were done on an electric analoge, others on a specially prepared laboratory stand. The results of the tests have been listed and discussed, on the basis of which some practical conclusions were drawn.

INTRODUCTION

Pressure pulsations of high amplitudes occur in the reciprocating compressor installations as a result of the cyclic work of these machines. Excessive pressure pulsations effect unfavourably both the work of the compressors and the compressed gas installations. In order to eliminate such negative effects the resonance phenomena in the installations should be prevented and the absolute values of pressure pulsations should be reduced. The reduction of the absolute values of pressure pulsations can be achieved by a proper configuration and dimensioning of the installation and a proper choice and location of pressure pulsation dampers [1], [4], [5].

PRESSURE PULSATION DAMPERS

For pressure pulsation damping in the compressed gas installations the pulsation dampers are applied. There are various designs of many types of dampers and patents attached but they exist with no full scientific grouds for their development and application. That is why many disappointments arise when the dampers which operate efficiently with one installation are completely unreliable and have the oppcsite effect in a case of some other installations. A damping effect much depends on a location of the damper in the installation as well as a characteristic impedance of the technological elements and fittings incorporated in the pipeline beyond the damper.

A proper selection of a pressure pulsation damper for a given installation is conditioned by taking into account the influence of all those parameters upon the operation efficiency of the damper. Additionally, the considerations should include the influence of the damper itself upon pressure pulsation rune both before and beyond it because the damper constitutes a certain concentrated inclusion in a homogeneous pipeline section. Such a complex examination of a pressure pulsation damper selection for a reciprocating compressor installation requires solving the partial differential equations which describe a pulsating gas flow along with the boundary conditions adequate for the examined installation. This task may be solved eithes by means of the electro-acoustic analogies using an analog or by the numerical methods on a digital computer [2], [3], [5].

A damper designed for the elimination of the excessive pressure pulsations in the reciprocating compressor pipelines should meet the following requirements:

- simple design, ease of making and assembling
- small overall dimensions
- reduction of pressure pulsation absolute values as much as possible
- small hydraulic resistance
- no harmful reverse effect upon compressor work.

A coefficient of pressure fluctuations in the pipeline behind the damper must not exceed the quantities acknowledged permissible.

The presented requirements are well satisfied by chamber damper single chamber and two-chamber resonance damper with a perforated central pipe, and a resonance chamber damper /Fig.1/. These types of dampers are recommended by the author to be applied for pressure pulsation damping in the gas compressor station installations.

DAMPING EFFECT EVALUATION CRITERIA

The measure of pressure fluctuation in the pipeline is a coefficient of pressure fluctuation determined as a ratic of a pressure pulsation absolute value Δp_a and mean pressure p_o in the pipeline

$$\delta = \frac{\Delta p_a}{p_o} \qquad /1/$$

To evaluate the effect of pressure pulsation reduction by means of a damper it is necessary to find a criterion by which this effect may be expressed in a numerical form. Such a criterion should also be a basis for comparing the effects of operation of various dampers in a given installation as well as indicate which damper is the best to be used under the given conditions.

Damping Ratio.

Most frequently used for measuring the effect of pressure pulsation reduction in the pipeline is the damping ratio K_t described as the ratio of pressure pulsation coefficients in the pipeline before and behind the damper.

$$K_t = \frac{\delta_1}{\delta_2} \qquad /2/$$

In case of a damper with small hydraulic resistance, the damping ratio is described as the ratio of pressure, pulsation absolute values before and behind the damper /Fig. 2/.

$$K_t = \frac{\Delta p_{a1}}{\Delta p_{a2}} \qquad /3/$$

A damping ratio defined in this way is convenient to use in experimental testing of dampers to compare their construction and evaluate the influence of different structural elements on a degree of pressure pulsation reduction. It shoud be remembered that damping ratio is characteristic of damping effect of dampers merely under certain idealised conditions, extremely rare in practice. It is assumed that in the pipeline sections before and behind the damper only the progressive pressure waves of constant amplitudes are propagated /Fig.2/.

Total Rate of Pressure Pulsation Damping.

The most general evaluation criterion of damper effective operation is the total rate of pressure pulsation damping K_g which determines lowering of a general pressure pulsation level along the whole length of piping. It includes the effect of reducing the pressure pulsation absolute values both before and behind the damper [1]. If Δp_{am} denotes a maximum pressure pulsation absolute values along the pipeline in the installation without damper /Fig.3a/ and this maximum is denoted as Δp_{amt} after having installed the damper /Fig.3b/, then the total rate of pressure pulsation damping can be defined as the ratio of those maximum values.

$$K_g = \frac{\Delta p_{am}}{\Delta p_{amt}} \qquad \text{/4/}$$

The evaluation of damper effective operation by means of a total rate of pressure pulsation damping K_g must be based on the knowledge of how pressure pulsation absolute values run along the pipeline. First, pressure pulsation absolute values along the pipeline without damper should be calculated including all the installed elements of equipment and fittings /coolers, oil separators, valves, etc./. Next, the analogical pulsation can be used after developing a way of calculating the pulsation absolute values along the pipeline including the installed elements. The total rate of pressure pulsation damping illustrates fully the reduction of the general level of pressure pulsations in the installation owing to the use of damper.

The Aim and Methodology of Research.

The aim of the experiments was to define the effect of the location and the inside structure of a damper on the efficiency of pressure pulsation damping in a discharge pipeline and on the work of a reciprocating compressor.

The experiments were carried out on a discharge pipeline of a laboratory compressed air installation and on its equivalent electrical model. The air was pressed by a single stage, single cylinder, single acting air compressor with the cylinder of 160 [mm] diameter and piston stroke of 150 [mm] . The compressor was driven by a d.c. electrical motor whose rotational speed was regulated steplessly in the range 200 - 1500 [rev/min] . Fig. 4 presents a three-dimensional diagram of the discharge pipeline with the given dimensions. The discharge pipeline 81,5 [mm] in diameter and 10,2 [m] length was terminated by a severely throttled valve. It was assumed that such a terminal meets the conditions of complete acoustic closure /$Z = \infty$/.

The experiments on the effect of the damper location on the efficiency of pressure pulsation damping in a discharge pipeline and on the work of a reciprocating compressor were carried out on an electrical model of a discharge pipeline. The efficiency of the damper operation was measured by total rate of pressure pulsation damping K_g, and the effect of the damper on the compressor work was measured by the absolute value of pressure pulsation Δp_a in the pipeline behind the compressor.

The experiments on the effect of the inside structure of a damper on the efficiency of pressure pulsation damping in a discharge pipeline and on the work of a reciprocating compressor were carried out on a laboratory stand. The efficiency of the damper operation was evaluated by means of damping ratio K_t, and the effect of the damper on the compressor work by means of the power delivered to the electrical driving motor N_{el} and the overall volumetric efficiency λ, which was defined as

$$\lambda = \frac{\dot{m}}{\dot{m}_t} = \frac{\dot{m}}{V_s\ \dot{n}\ 60\ \rho_d} \qquad \text{/5/}$$

where: $\dot{m}$ - the real delivery of the compressor measured in a discharge pipeline, $\dot{m}_t$ - theoretical delivery of the compressor, V_s - swept volume of cylinder, $\dot{n}$ - rotational speed of the compressor shaft, ρ_d - density of suction medium.

THE RESULTS OF EXPERIMENTS

The Effect od Pressure Pulsation Damper Location.

The experiments on the effect of the location of a damper on the efficiency of pressure pulsation damping in a discharge pipeline and on the work of a reciprocating compressor were carried out at constant absolute pressure of 3[bar] of the outlet and rotational speeds varying in the range 500 - 1300 [rev/min] . Below are presented some examples of the results of the tests in case when a chamber damper was used for pressure pulsation damping.

Fig. 5 and 6 show in diagrams of formula $\Delta p_a = f/l/$ the results of tests on the effect of location of a chamber damper on its efficiency at rotational speed of 500 and 900 [rev/min]. The chamber damper of 0,0956 [m^3] of volume was placed at the distance of 0,3, 1,875 and 5,325 [m] from the compressor. In Table 1 are shown, for both rotational speeds and particular distances, absolute values of pressure pulsation Δp_a in the pipeline behind the compressor as well as the values of total rate of pressure pulsation damping K_g and maximum coefficient of pressure fluctuation δ_m in the pipeline behind the compressor.

Table 1

Distance between chamber damper and compressor [m]	Δp_a [bar]	K_g	δ_m
500 [rev/min]			
0,3	0,19	3,00	0,027
1,875	0,355	1,61	0,032
5,325	1,58	0,36	0,197
900 [rev/min]			
0,3	0,34	1,53	0,033
1,875	0,56	0,93	0,053
5,325	1,09	0,48	0,08

From the diagrams of formula $\Delta p_a = f/l/$ presented in Fig. 5 and 6 the significant effect of the location of a chamber damper on the efficiency of its operation becomes evident. Placing it at the distance of 5,325 [m] from the compressor caused a significant rise in pressure pulsation absolute values immediately behind the compressor in comparison with those in the pipeline without a damper. The effect of a chamber damper location on the pressure pulsation rate numerically characterizes the total rate of pressure pulsation damping. The smaller the distance between the compressor and damper, the weaker the influence of air pulsating stream on the compressor work and lower the pressure pulsation absolute values in the pipeline behind the damper. When this distance is longer, the pressure pulsation absolute values on the compressor-damper and behind the damper sections rise.

The Effect of the Pressure Pulsation Damper Inside Structure.

The investigation on the effect of the damper inside structure on its damping efficiency in a discharge pipeline and on the work of a reciprocating compressor was carried out at the constant absolute pressure of 3 [bar] of the outlet and rotational speeds between 500 and 1300 [rev/min] . A single chamber damper and double chamber resonance dampers with perforated central pipe were tested as well as resonance chamber damper and chamber resonance damper. In the double chamber resonance damper the proportion of chamber volumens was changed. All the tested dampers had the same inside diameter of 0,499 [m] and length of 0,604 [m] . The perforations of the central pipeline of the resonance dampers were 11 [mm] in diameter. The dampers were placed at 0,3 [m] from the compressor.

On the basis of the experiments it was found that the change of the inside structure of the damper placed in a discharge pipeline at 0,3 [m] from the compressor does not practically affect the overall volumetric efficiency λ . The effect of the damper inside structure on the electric power N_{el} required for compressing and damping ratio was shown in Fig. 7 and 8 as diagrams of formulae $N_{el} = f/\dot{n}/$ and $K_t = f/\dot{n}/$. From the diagrams it is evident that the inside structure of the damper placed at this distance from the compressor affects significantly the damping ratio and has a negligible effect on the electrical power necessary for compressing. On the whole, the best results were obtained in case of a single chamber resonance damper.

CONCLUSIONS

On the basis of the investigation it was found that the location of a damper in a discharge pipeline has an essential effect on the efficiency of pressure pulsation damping and the work of a compressor. The inside structure of a damper placed at an optimum point of a discharge pipeline does not have any practical effect on the volumetric efficiency, a minor effect on the power for compression and a significant effect on the damping ratio. By changing the inside structure of a damper the damping of particular harmonic components of a pulsating gas flow can be affected.

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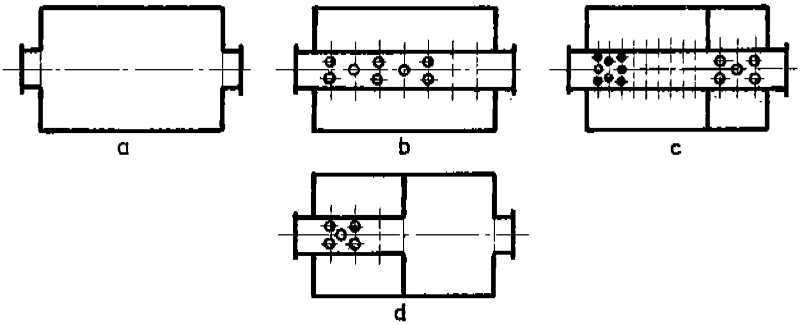

Fig. 1. Pressure pulsation dampers: a - chamber, b - single-chamber resonance, c - two-chamber resonance, d - resonance-chamber.

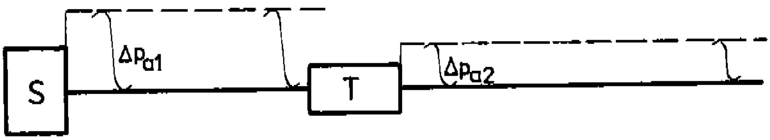

Fig. 2. Diagram to determine damping rate: S - compressor, T - damper.

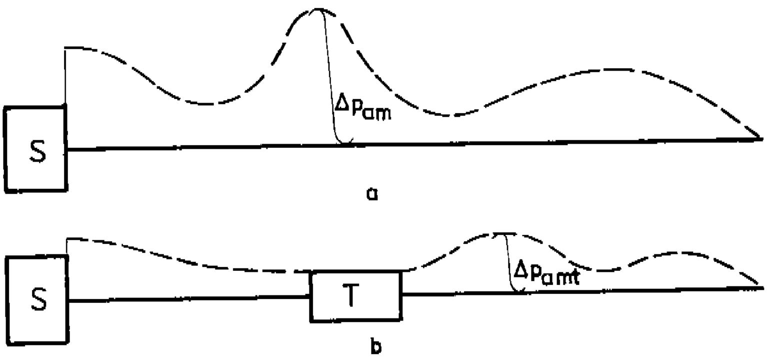

Fig. 3. Diagram to determine the total rate of pressure pulsations damping.

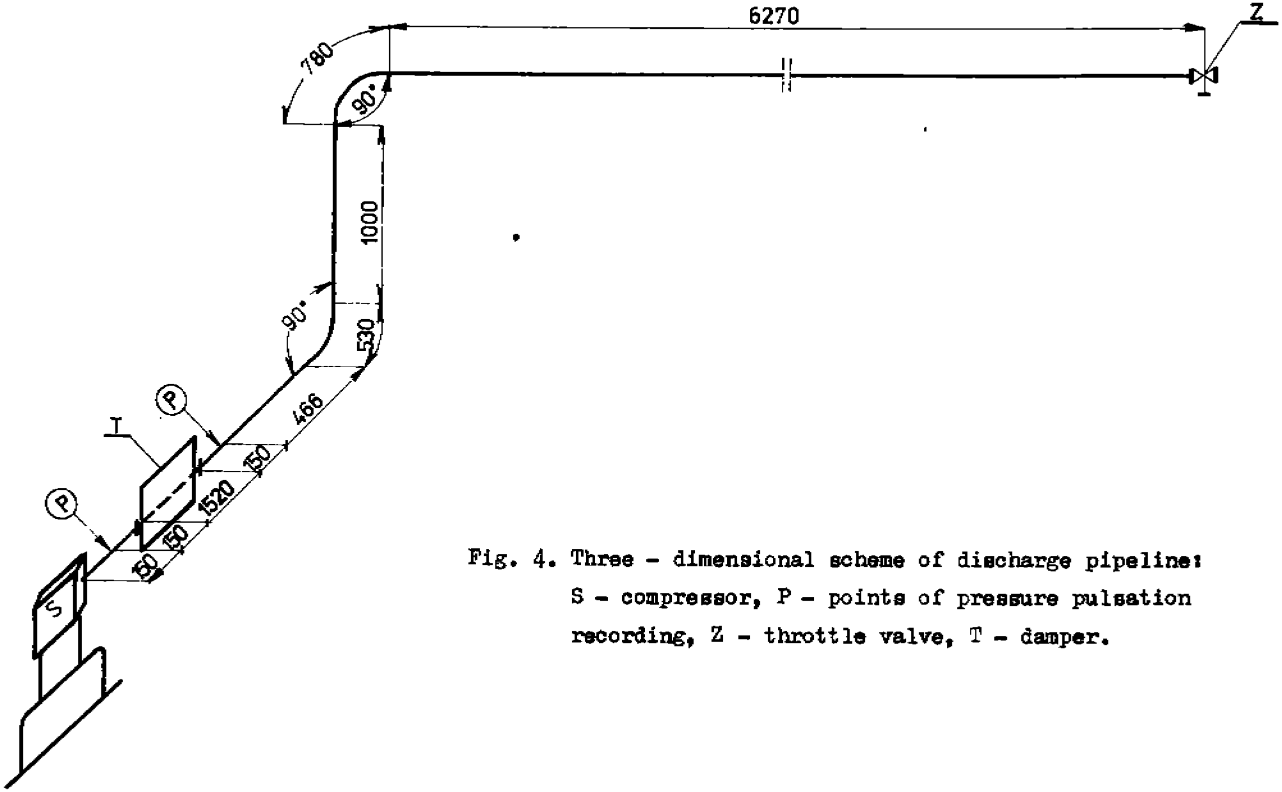

Fig. 4. Three - dimensional scheme of discharge pipeline: S - compressor, P - points of pressure pulsation recording, Z - throttle valve, T - damper.

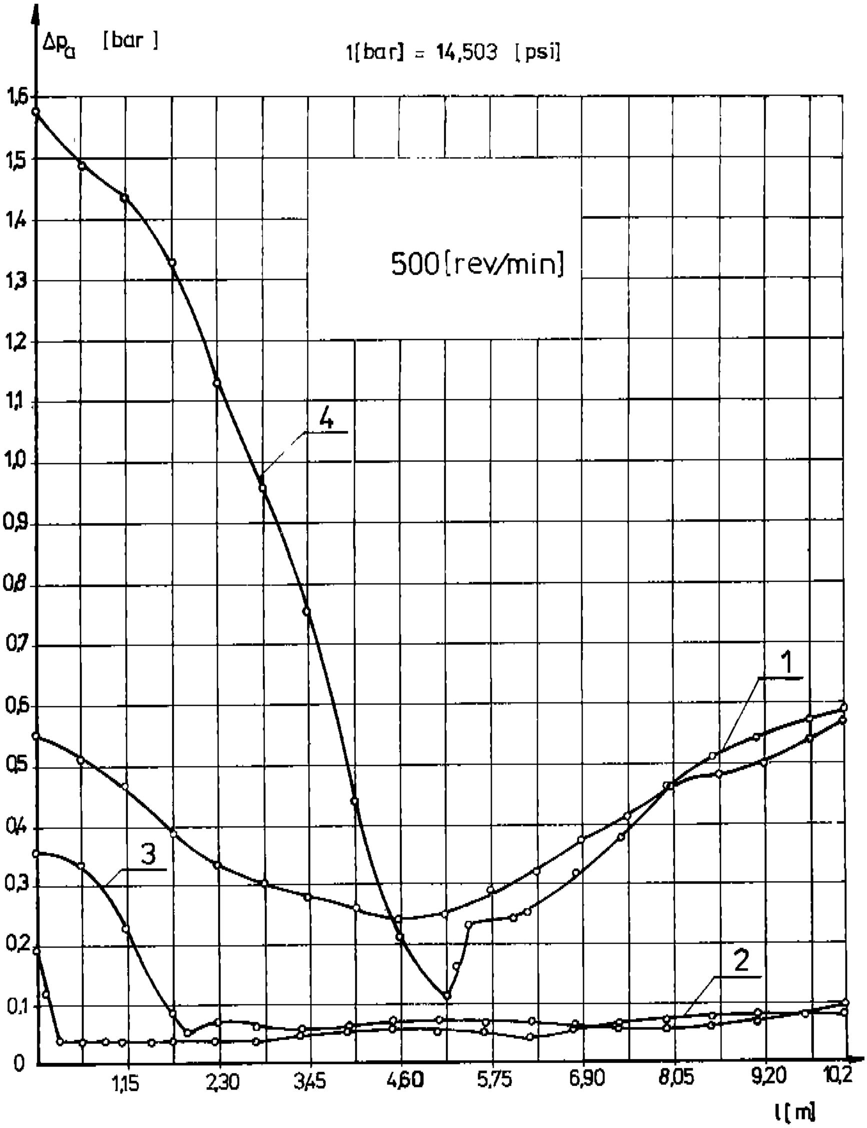

Fig. 5. Diagrams of $\Delta p = f/l/$ for different distances of chamber damper from compressor at 500 [rev/min]: 1 - outlet pipeline without chamber damper, 2 - chamber damper at 0,3 [m], 3 - chamber damper at 1,875 [m], 4 - chamber damper at 5,325 [m].

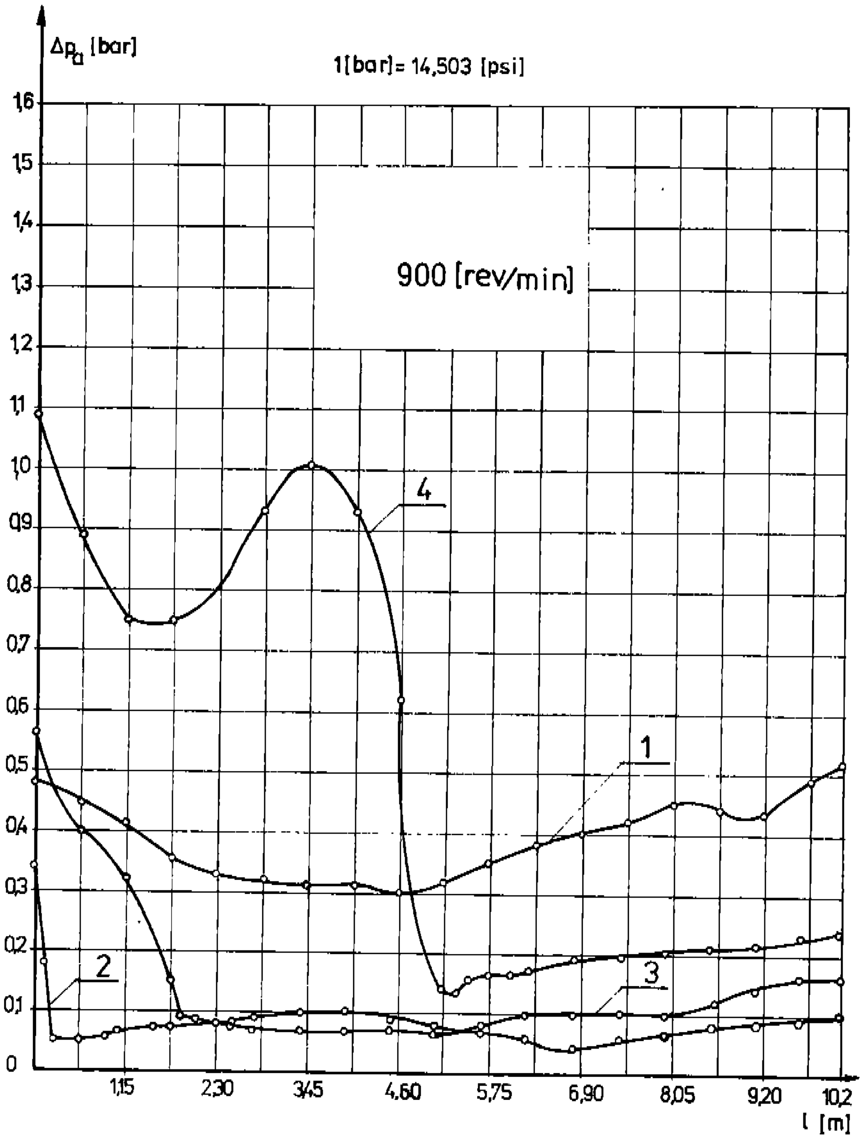

Fig. 6. Diagrams of $\Delta p = f/l/$ for different distances of chamber damper from compressor at 900 [rev/min]: 1 – outlet pipeline without chamber damper, 2 – chamber damper at 0,3 [m] , 3 – chamber damper at 1,875 [m] , 4 – chamber damper at 5,325 [m].

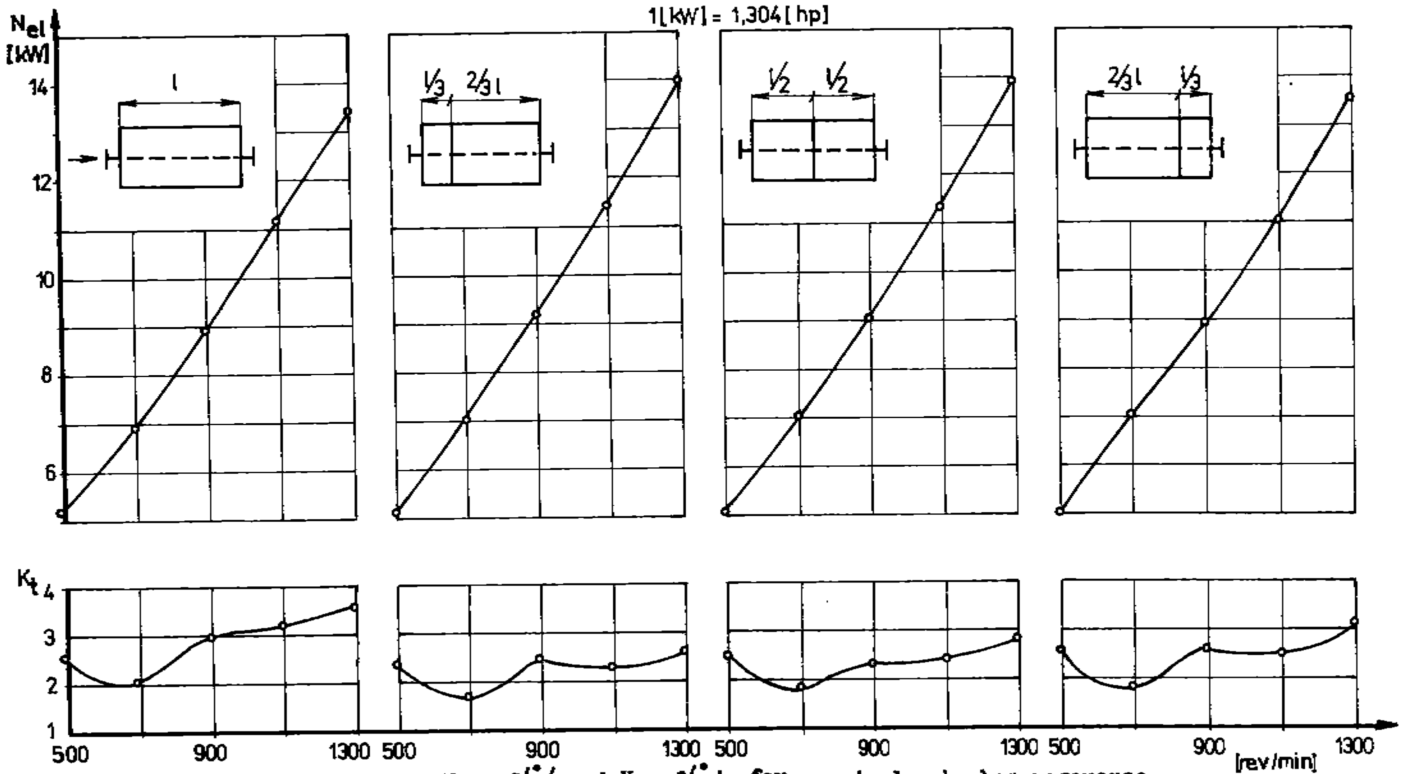

Fig. 7. Diagrams of formulae $N_{el} = f/\dot{n}/$ and $K_t = f/\dot{n}/$ for a single chamber resonance damper and double chamber resonance dampers.

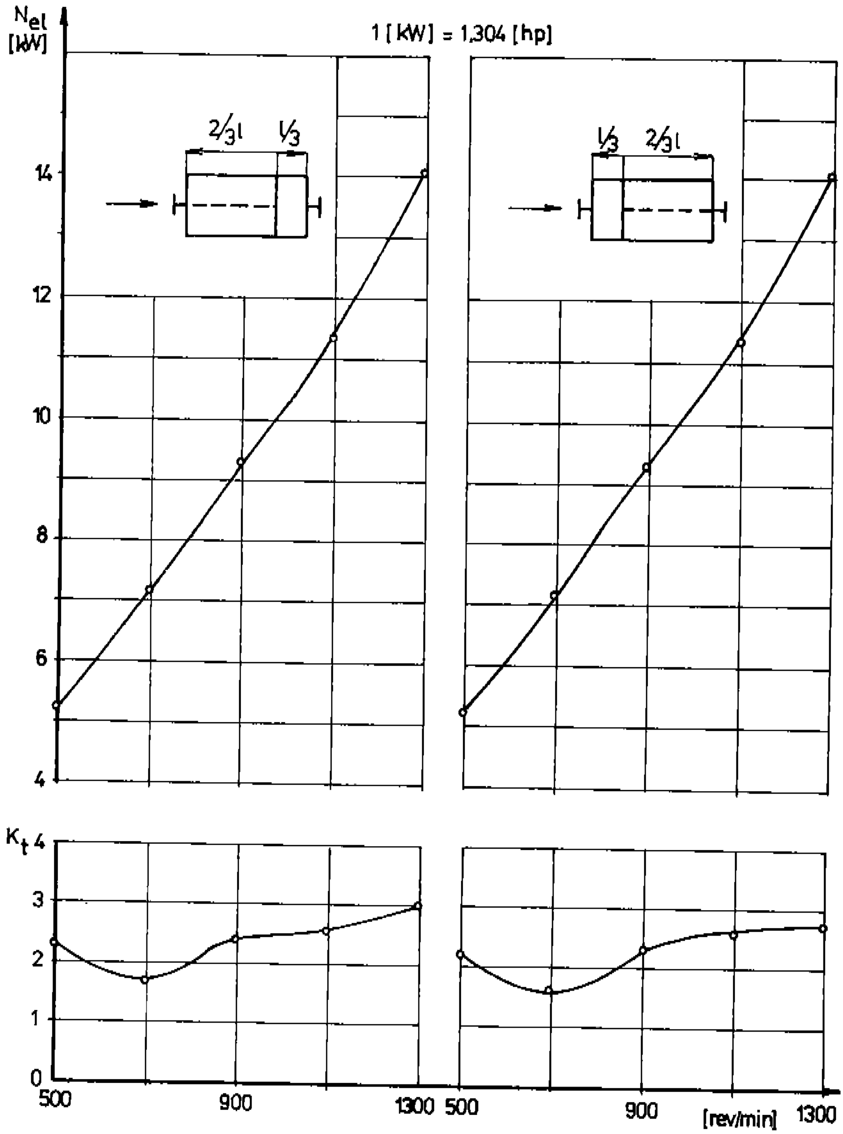

Fig. 8. Diagrams of formulae $N_{el} = f/\dot{n}/$ and $K_t = f/\dot{n}/$ for a resonance chamber damper and a chamber resonance damper.