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# CALCULATION AND DESIGN OF FLYWHEEL FOR SMALL AND MEDIUM COMPRESSORS

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## I Introduction

Nowadays most of the machine parts respectively complete devices are often examined to save production costs. The purpose of this paper is to provide data for an easy calculation of the necessary flywheel dimension. The weight of the flywheel usually is a considerable amount of the total weight of a reciprocating compressor. Saving weight of the flywheel cuts down the production costs.

## II The coefficient of cyclic variation $\delta$

The flywheel of a reciprocating compressor keeps the variations of the cyclic motion in limits. To state the irregularities of the cyclic motion the coefficient of cyclic variation is defined as  $\delta$ , fig. 1. By an adequate design of the flywheel  $\delta$  can be kept in limits. For small and medium compressors the following values can be applied [1], [2]

$\delta = 1/30$  to  $1/40$  for electric motor driven compressors (belt)

$\delta = 1/75$  to  $1/100$  for electric motor driven compressors (direct drive with elastic coupling)

$\delta = 1/10$  for Diesel driven compressors (compressor-motor bloc device) and lowest permitted speed. With standard speed  $\delta$  is considerably lower, about  $1/40$  to  $1/60$

An AC motor cannot endure too great variations of cyclic motion as they occur in de-

vice with small flywheels. This also refers to elastic couplings. The variation of cyclic motion can lead to torsional vibrations of the shafting (consisting of crankshaft, flywheel, rotor of the AC motor and elastic coupling).

As the exact calculation of the flywheel is very extensive, one does it often by estimation. In this case the flywheels are then mostly bigger than required. A big flywheel has of course serious disadvantages. There is f.i. no advantage in having a device with  $\delta=1/200$  instead of  $\delta=1/100$ . The disadvantages are the following:

1. A double-sized flywheel ( $\delta=1/200$ ) warms the windings of the AC motor up to the double amount at each start
2. The heavy flywheel ( $\delta=1/200$ ) charges the bearings and the crankshaft more intensively than a light one ( $\delta=1/100$ ), not only by its own weight but also by centrifugal forces, especially at high speeds. This is because a heavy rotor has a bigger residual unbalance after balancing.
3. A bigger flywheel makes the compressor heavier and more expensive.
4. Under extreme circumstances a flywheel which is designed too big can lead to torsional vibrations of the crankshaft or shafting, followed by fractures.

## III Rough calculation of flywheel

A rough calculation of the flywheel can be done easily by equation (1) [3].

$$J = k \frac{P_i}{\delta \cdot n^3} \quad (1)$$

with  $J \dots \text{kg} \cdot \text{m}^2$  moment of inertia of flywheel (SI units !)  
 $P_i \dots \text{kW}$  Power indicated of compressor  
 $n \dots \text{rpm}$  speed  
 $k \dots \text{kg} \cdot \text{m}^2 / \text{kW} \cdot \text{min}^3$  constant factor

Equ(1) is widely used for the calculation of the flywheel of IC engines and one can find the corresponding  $k$  values in the manuals. Now the authors have calculated  $k$ -values for compressors, i.e. for AC motor driven one- and two stage-compressors and for Diesel driven compressors (motor and compressor working with one crankshaft). The  $k$  values were calculated by computer, and found by comparison with an exact calculation [3].

The calculated  $k$  values are compiled in table 1 (electric driven or driven by motors with uniform torque at coupling) and table 2 (driven by a 4-stroke Diesel engine).

It is interesting to compare cylinder arrangements of 2-stage V- and W-type devices according table 1b and 1c. For the V-device the  $k$  value (as well as the required flywheel size) depends considerably on the direction of the rotation (difference about 20% !). Also for the W-arrangement with the 2<sup>nd</sup> stage in the center has disadvantages concerning the flywheel. Such a device requests a flywheel bigger by 30%, compared to the more favorable asymmetric arrangement (whereby the direction of rotation is important !). One can easily explain the reason for the difference in the  $k$  values for the V-device: when the 2nd stage has the suction period, the incoming gas does work and such an arrangement will be favorable which can use this work for compression in the first stage. With the W-type engine this is similar.

For the calculation of the flywheel according equ(1) one needs the indicated power  $P_i$  of the compressor. This can be calculated from the power consumption  $P_e$  and from an estimated mechanical efficiency  $\eta_{\text{mech}}$  according equ(2)

$$P_i = P_e \cdot \eta_{\text{mech}} \quad (2)$$

For the exact calculation a number of relatively unimportant parameters is necessary. For the rough calculation according equ(1) one only needs the indicated power  $P_i$  and the speed  $n$ . The parameters used with the computer program were the following:

polytropic coefficient 1.35 for compr.  
 1.20 for return motion  
 clearance Volume 5%  
 ratio crankshaft radius/ connecting rod length 1/4  
 steps/round: 60 i.e. 6<sup>o</sup> steps  
 suction pressure (cylinder): 0.9 bar  
 delivery pressure (cylinder):  $1.08 \cdot p_2/p_1$   
 2nd stage: similar as in 1st stage  
 mass forces according to a pressure of 1.75 bar in upper dead center  
 Diesel engine: standard indicator diagram with 75 bar maximum pressure

Friction was not taken into consideration. Only the variable amount of friction has an influence (piston friction). The constant amount of friction does not count for the flywheel. There is of course a weak influence on the  $k$  value caused by the speed (via inertia forces of piston). But this influence is very small as a check with the computer showed. Also the influence of the clearance volume is very small, provided that the clearance volume is within the limits of 3-7%.

The rough calculation according equ(1) is now demonstrated in the following example: For a 2-stage W-type compressor with the given data a flywheel for  $\delta=1/100$  is to be designed.

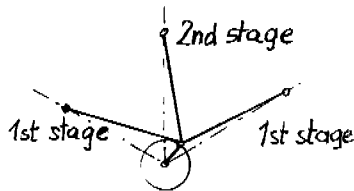
Data: the quantity delivered:  $5.8\text{m}^3/\text{min}$   
 pressure: 9 bar(abs)  
 power consumption  $P_e=38.4\text{kW}$   
 speed  $n=1500\text{ rpm}$

According to the size of the machine we estimate the mechanical efficiency to  $\eta_{\text{mech}}=0.9$  this gives

$$P_i = 0.9 \times 38.4 = 34.6 \text{ kW}$$

We compare 2 cylinder arrangements: one arrangement with the 2nd stage in the center (symmetric) and this asymmetric arrangement with the most favourable k value (see table 1)

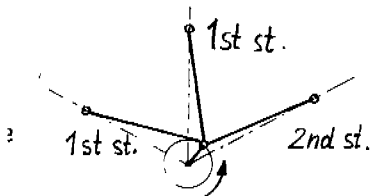
a) symmetric arrangement



$$k = 3.92 \times 10^6 \text{ kg.m}^2/\text{kW.min}^3$$

$$J = k P_i / \omega n^3 = 4.03 \text{ kg.m}^2$$

b) asymmetric arrangement



$$k = 2.98 \times 10^6 \text{ kg.m}^2/\text{kW.min}^3$$

$$J = 3.06 \text{ kg.m}^2$$

It should be mentioned that the still used "flywheel effect  $WR^2$ " ( $\text{kp.m}^2$ ) is exactly four times as big as the inertia moment ( $\text{kg.m}^2$ ) i.e.  $1\text{kg.m}^2$  corresponds to  $4\text{kp.m}^2$ .

Usually one only takes the flywheel ring in account and demands  $J_{\text{ring}} = 0.9 \times J$  for a disc flywheel and  $J_{\text{ring}} = 0.95 \times J$  for a spoked flywheel. We choose a disc flywheel made of steel casting. For this material the permitted circumferencial speed amounts to

50 m/s. This gives a permissible outer diameter of  $D=0.63\text{ m}$  for the speed of 1500 rpm. The inertia moment of a hollow cylinder with density  $\rho$  (notations according to fig. 2) is

$$J = \frac{1}{2} \rho b (D^4 - d^4)$$

When we choose  $D=0.6\text{m}$  and use  $\rho=7300\text{kg/m}^3$  for casting we get

a) symmetric arrangement | b) asymmetric arr.

$$b = 7.5\text{cm}$$

$$m = 47.4 \text{ kg}$$

$$b = 5.5\text{cm}$$

$$m = 36 \text{ kg}$$

considering the fact that the ring needs only 90% of the calculated inertia moment. For the asymmetric arrangement we save 11.4 kg casting without any change in the coefficient of cyclic variation.

The computer program fits for any cylinder and stage arrangement. But it is only reasonable to table k values for widespread types. For special types separate calculations must be done. The computer program is also adaptable for piston pumps.

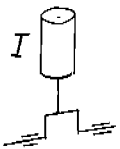
#### Literature

- [1] FRENKEL, M.I.: "Kolbenverdichter" Berlin, 1969.
- [2] KÜTTNER, K.-H.: "Kolbenmaschinen" Stuttgart, 1967.
- [3] BÖSWIRTH, L.: "Das Schwungrad kleiner und mittlerer Kompressoren". Journal "druckluft-praxis" 5(1973), Heft 4. Mainz.

TABLE 1

k-values for the calculation of the flywheel. Electric driven compressors or compressors driven by motors with uniform torque at coupling. k in kg.m<sup>2</sup>/kW.min. Suction pressure lbar.

Ⓐ parallel cylinders

Cylinder- and stage arrangement				2 cylinders 1st stage			2 cylinders 2 stages		
	pressure bar(abs) (container)	7	9	11	7	9	11	7	9
$10^{-6}k$	3.81	4.20	4.65	0.81	1.05	1.29	2.32	2.18	2.12

Ⓑ V-type compressors (V-angle 90°)

Cylinder- and stage arrangement	1 stage			2 stages			2 stages		
	pressure bar(abs)	7	9	11	7	9	11	7	9
$10^{-6}k$	2.96	3.14	3.34	3.46	3.37	3.32	4.13	4.09	4.10

Ⓒ W-type compressors (W-angle 2x 60°)

Cylinder- and stage arrangement	1 stage			2 stages symmetric			2 stages asymmetric	
	pressure bar(abs)	7	9	11	7	9	11	9
$10^{-6}k$	2.54	2.68	2.84	4.01	3.92	3.91	3.74	2.98

Ⓓ double V- and W-type compressors

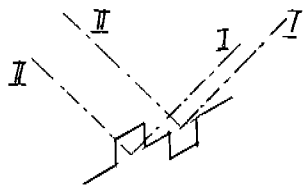
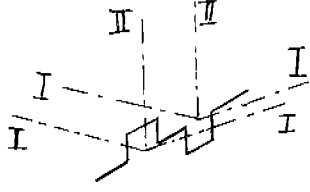
Cylinder and stage arrangement						
	pressure bar(abs)	7	9	11	9	
$10^{-6}k$	0.28	0.31	0.48	0.48		

TABLE 2  
 k-values(kg.m<sup>2</sup>/kW.min<sup>3</sup>) Diesel driven compressors. 4-stroke Diesel engine and compressor working at the same crankshaft. Suction pressure 1 bar(0.9bar in cylinder). I...1st stage, II...2nd stage, D...Diesel cylinder. D(90°) ... means f.i.: injektion 90° after drawn pos.

a parallel cylinders, 2 stages

Cylinder and stage arrangement			
pressure bar(abs)	9	9	
10 <sup>-6</sup> k	9.76	8.33	

b V-type engine, 1cyl.Diesel, 1cyl.compr.

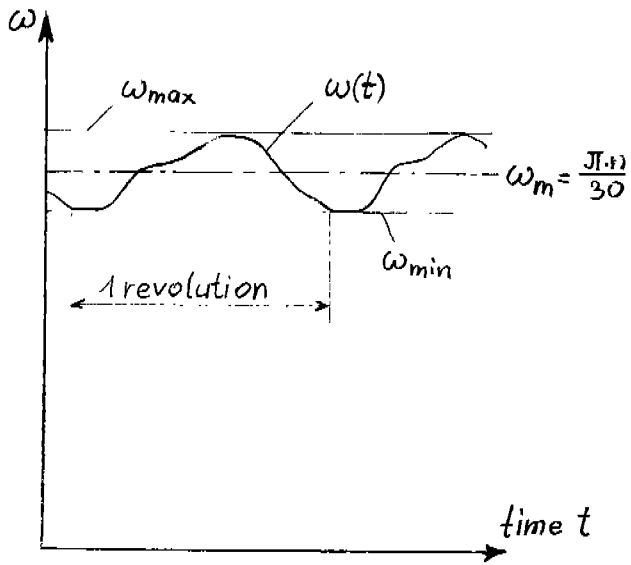
Cylinder and stage arrangement			
pressure bar(abs)	4	7	9
10 <sup>-6</sup> k	9.91	9.75	9.71
	14.39	14.37	14.49

c double V-type engines

Cylinder and stage arrangement			
pressure bar(abs)	5	9	9
10 <sup>-6</sup> k	6.51	6.55	6.52

d double V-type engines

Cylinder and stage arrangement			
pressure bar(abs)	7	9	9
10 <sup>-6</sup> k	7.50	8.10	7.37



$$\delta = \frac{\omega_{max} - \omega_{min}}{\omega_m}$$

Figure 1

Definition of the coefficient of cyclic variation.  $\omega$ : angular speed.

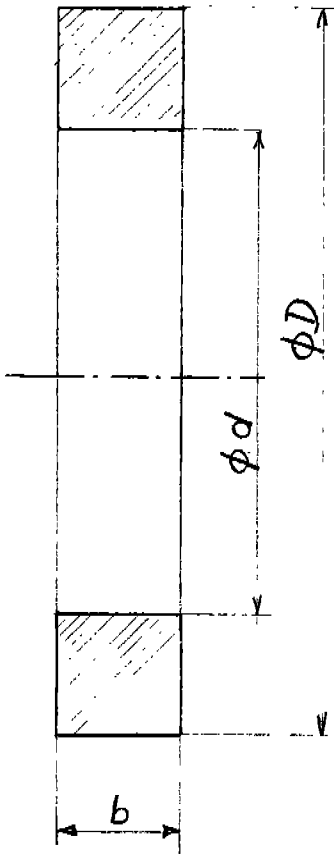


Figure 2

Flywheel dimensions, notation.