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An Approach to Reckoning of Peak Noise Frequency of Air Compressor Inlet

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AN APPROACH TO RECKONING OF PEAK NOISE FREQUENCY OF AIR COMPRESSORE INLET

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ABSTRACT

The peak noise frequency of air compressor inlet is important parameter when designing suction muffler of air compressor. From the aeroacousties this paper has derived the reckoning formula for peak noise frequency of air compressor inlet. Through the formula the peak number of noise frequency and peak frequency of air compressor inlet can be calculated. This paper nadverified the reliability of the reckoning formula with spectrum of eight different air compressor, the result of noise capacity of which is 0.6-40m³/min and the calculation agree well with the result . The formula will render some convenience for designing suction muffler of air compresser.

SYMBOLS

C	piston instantaneous velocity	m/s
D_h	piston diameter	m
F_g	thoroughface area of suction pipe	m^2
F_h	piston area	m^2
F_f	thoroughface of valve clearance	m^2
K	coefficient of compressor single and double acting	
N	number of the first grade suction valve that work at the same time	
S	Strouhal's number	
V	airflow velocity	m/s
V_f	airflow velocity of valve clearance	m/s
V_g	airflow velocity of suction pipe	m/s
d	equivalent jet diameter	m/s
f_m	peak frequency of pulsating airflow noise	Hz

f_g	peak frequency of valve jet noise	Hz
h	valve-flake rising-distance	m
m	clearance expand index	
n	air compressor speed	r.p.m.
r.	radius of crant throm	m
u.	piston mean velocity	m/s
α	the first grade relatively clearance volume	0/0
	rotative angle of crankshaft	degree
θ	suction valve closing angle of the first grade	
θ_h	suction valve closing angle of crankshaft-side	
θ_l	suction valve closing angle of lid-side	
ε_1	the first grade compress proportion	
λ	connecityying red proportion	
λv	volume coefficient of the first grade	0/0
ω	compressor angular velocity	
β	suction valve flow coefficient of the first grade	

INTRODUCTION

It is know to all that the noise of air compressor inlet is main noise source of air compressor. A suction muffler is an important means to air compressor noise control. Therefore suction muffler designing has become a froemost reseach problem of air compressor noise control. When we begin to design a sucton muffler, the first question how to define the peak noise frequency of air compressor inlet according to the known air compressor parameter, this peak frequency is the main basis for the designing of the muffler structure and for the choosing of a sound absorption material for the muffler, it the structure of the is different, the peak noise frequency of air compressor inlet is different, too.

There are some inlet noise spectrums of the air compressor in Fig 1, which are common in China. We can discover some problems from the figures. Why there is only one peak in some sound spectrums and there are two peak in the others? Why some peaks frequency have 125 Hz octave and other have 250 Hz octave. For instance type 2Z-3/8, V3/8-1, 3L-10/8 air compressors have one peak, and type 2V-0.6/7, L2-1 0/8-1, L3.5-20/7,L5.5-40/8 air compressors have two peaks. Why type 2V-0.6/7, 2Z-3/8-1, L2-10/8 air compressors have peak in the 250 Hz octave and type V-3/8-1, 3L-10/8, L3-10/8, L3.5-20/7, L3.5I-20/8, L5.5-40/8 air compressors have peak in the 125 Hz octave. Are their peak frequencies a law? And how do we look for the law by the known parameters of the air compressor. To grasp the law of the peak frequencies are of much help to us for developing of the a new air compressor.

The distinction of this paper from other similer paper is biffish. The derivative basis of this paper is the valve jet aero-thermo-acoustics. To use the derivave reckoning formula is very easy and there is no need for computer simulation.

DERIVATION OF THE RECKON FORMULA

Noise[1] of the air compressor inlet is the piling up[2] of every noise which can be produced from the air compressor inlet to suctionsystem confining surface piston top. The noise of the air compressor inlet consists of the noise suction valve jet, the noise of the airflow of the suction pipe, the impact noise of the piston ring, impact noise of the suction and valve plate, freation noise of the cylinder wall. The noise frequency of the air compressor inlet is composed of these, too. We had replaced the original valve plate and piston ring by some new damping material, so that we could find the biggest source among them.

The results tell us that the noise of the suction valve jet and the pulsating airflow noise of the suction pipe are by far the largest of all sound in the suction system. So in this paper we hold that impact noise and the fraction noise should be neglegible.

So the jet noise of the suction valve and the pulsating airflow noise of the suction pipe are main noises sources of the air compressor inlet noise. And that is to say the peak frequency of the suction valve jet noise and the peak frequency of the pulsating airflow noise of the suction pipe are peak frequencies of the air compressor inlet noise.

Therefore to look for noise peak frequency of air compressor inlet is to calculate the suction valve noise peak frequency and the suction pipe pulsating airflow noise peak frequency.

1. The suction pipe pulsating airflow noise peak frequency can be calculated. On the basis of aeroacousties we know that the air is not only a medium of sound but also produce sound when the air is flowing. Under the given circumstances this air borne sound controls the velocity of airflow. When it is sucking, the air of the air compressor suction pipe has the velocity and can produce air-borne sound. Because the sucking of the piston air compressor is an intermittence, So the airflow velocity of the suction pipe is an intermittence, too. The intermittence of the airflow velocity is identical with the compressor shaft rotational speed. Because the airflow velocity changes, so changes the air-borne sound and pulsating. And the frequency of air-borne change is identical with the frequent of the airflow velocity change. That is to say the

peak frequency of the airflow velocity change.

To look for the peak frequency of the airflow velocity change is our aim. With this aim in view we start to look for the peak frequency of the airflow velocity change. Based on the law of continuity we can write the airflow velocity V_p of the suction pipe as follows:

where $$V_p = \frac{Fc.\ C}{F_P}$$

$$C = \gamma.\omega\ .(\sin\alpha + \frac{\lambda}{2}\sin 2\alpha)$$

so $$V_p = \frac{\gamma\cdot\omega\cdot Fc}{Fp}(\sin\alpha + \frac{\lambda}{2}\sin 2\alpha) \qquad ...(1)$$

In one period of the shaft rotational, some times the suction valve is closing and others opening. If the crank angle degree at starting clearance volume expansion is 0 . The crank angle degree at starting suction is θh or θd, so the functional equation of the airflow velocity of the suction pipe is single acting.

$$V_p = \begin{cases} 0 & 0^\circ \le \alpha \le \theta \\ \dfrac{\nu\cdot\omega\cdot Fc}{Fp}(\sin\alpha + \frac{\lambda}{2}\sin 2\alpha) & \theta < \alpha \le 180^\circ \\ 0 & 180^\circ < \alpha \le 360^\circ \end{cases} \quad ...(2)$$

double acting

$$V_p = \begin{cases} 0 & 0^\circ \le \alpha \le \theta h \\ \dfrac{\nu\cdot\omega\cdot Fc}{Fp}(\sin\alpha + \frac{\lambda}{2}\sin 2\alpha) & \theta h < \alpha \le 180^\circ \\ 0 & 180^\circ < \alpha \le \theta d \\ \dfrac{\nu\cdot\omega\cdot Fc}{Fp}(\sin\alpha + \frac{\lambda}{2}\sin 2\alpha) & \theta d < \alpha \le 360^\circ \end{cases} \quad ...(3)$$

Based on the formula (2), (3) we see the velocity V_p isn't elemetary function.Their functional curves can be seen in Fig.2, we see that they aren't simple harmonic function.

From above we see these functions are periodic functions. So we trandsform them into Fuller's functions with ease in looking for the biggest harmonic. This harmonic functional frequency is the peak frequency of the air-flow velocity V_p of the air compressor suction pipe.

where $$V_p = \frac{1}{2}a_0 + \sum_{n=1}^{\infty}(a_n \cos n\alpha + b_n \sin\alpha) \quad ...\ (4)$$

$$a_n = \frac{\gamma\cdot\omega\cdot Fc}{\pi\ \ Fp}\int_0^{2\pi}|V_p|\cos n\alpha\, d\alpha \qquad (n=0,1,2,...)$$

$$b_n = \frac{\gamma\cdot\omega\cdot Fc}{\pi\ \ Fp}\int_0^{2}|V_p|\sin n\alpha\, d\alpha \qquad (n=1,2,...)$$

Electronic compressor calculation tell us that biggest harmonic is 1st (the single acting compressor) or 2nd (the double acting compressor). Because the frequency of compreeor rotational speed times harmonic ordinal number is frequency of this harmonic.

So the peak frequency of the single acting compressor airflow velocity Vp is frequency of the compressor rotational speed. And the peak frequency of the double acting compressor airflow velocity Vp is twice as the frequency of the compressor rotational

means: $$f_m = \frac{k \cdot n}{60} \qquad \ldots(5)$$

single acting K=1; double acting K=2

2. The noise peak frequencies of suction valve jet when suction valve works and because the velocity of airflow through the valve clearance is quick, from the material [3], the suction valve will make jet noise sound spectrum is still continual broadband, from low frequency to high frequency, the compoments are rich, and certain frequency, there is a higher peak: it's spectrums of sound copacity is the function of Strouhal's number $\frac{fd}{V}$ [3]; it's peak sound spectrum will also change when jet velocity changes, but the peak frequency of this spectrum is always on the spectifical frequency f, whether it changes.

$$f = \frac{sV}{d} \qquad \ldots(6)$$

s=0.15-0.20

Passage of the air compressor valve general are perallel connected with many plays of valve, whether annual valve, straing-flow valve or other type valve, we all can found an equivalent throttle installation that were perallel connected with many little nozzles. As Fig.3 caption.

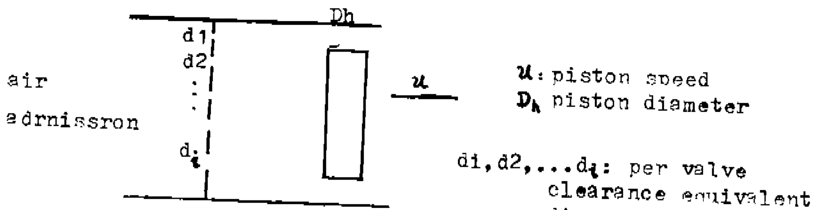

Fig.3 — suction valve equivalency

As airflow velocities are equality which per through parallel connection throttle installation, and equivalent diameters of every valve clearance in one's valve are different, from (6) seen: per clearance jet noise peek frequency is different. If a valve has i links, jet noise will have i peak frequencies.

$$f_{s1} = \frac{sV}{d_1}, \quad f_{s2} = \frac{sV}{d_2}, \ \ldots \ f_{si} = \frac{sV}{d_i}$$

So, we can see valve as a phonation-evement, which produce many frequencies tone at the same time. If airflow pass several holes at the same time, it can produce several frequencies tone. Low or high of valve jet noise frequency is decided by every valve clearance jet sound-power. The largest sound-power of clearance jet noise peak frequency is this valve peak frequency, for annualar valve, from Lighthill's law[3] know, in supplies condition, jet noise is directly proportional to jet diameter square. For the valve in working order, the most exterior annuallation-valve clearance is the largest, so it's jet noise is the largest than other valve clearance. So, it's peak frequency is peak frequency of valve jet noise. If medium-diameter of the most exterior annualation valve-flake is D1, valve-flake rising-distance is h, according to area conversion written it's jet equivalent diameter $d = \sqrt{8hD1}$ ...(7)

Airflow speed of valve clearance Vf through (1) formula may look for, exchange of Fg for Ff in the (1) formula, as it is, crank up then angle in θ-180° the suction valves, so airflow average speed of valve clearance Vf:

$$V_f = \frac{\gamma \cdot \omega \cdot F_h}{F_f} \cdot \frac{180}{\pi\,(180-\theta)} \int_{\theta}^{180} \left/ \sin\alpha + \frac{\lambda}{2}\sin 2\alpha \right/ d\alpha$$

angular velocity $\omega = \frac{\pi n}{30}$ and piston velocity $u = \frac{\gamma \cdot n}{15}$

Can be put in the mentioned formula, than it can be abtained

$$V_f = \frac{90 \cdot u \cdot F_h}{(180-\theta)F_f} \left[(1+\cos\theta) - \frac{\lambda}{4}(1-\cos 2\theta) \right] \quad \cdots (8)$$

uaually, the first grade compress proportion ε_1 is taken small, it's only 0.9-0.95[4] multiple of equal compress proportion designer in order to lessen measure and weight of compressor, in order to heigher volume of coefficient λ_v must restrict in the least clearance volume of the first grade, cause the first grade suction valve is closing early, mean θ is less, so the second item $\frac{\lambda}{4}(1-\cos 2\alpha)$ in bracket of aforecited formula is less than item infront, may neglect. So aforecited formula simplify

$$V_f = \frac{90 \cdot u \cdot F_h}{(180-\theta)\ F_f}\ (1+\cos\theta) \quad \ldots (9)$$

because closing angle of suction valve is:

$$\theta = \cos^{-1}\left[1 - 2\alpha\left(\varepsilon_1^{\frac{1}{m}} - 1\right)\right]$$

$$\lambda_v = 1 - \alpha\left(\varepsilon_1^{\frac{1}{m}} - 1\right) \quad \cdots (10)$$

When replacing the formula (10) in the above formula, it's possible to get closing angle θ which is expresser as the volume coefficient λ_v

$$\theta = \cos^{-1} (2\lambda_v - 1) \qquad \ldots(11)$$

replace into (9) with (11) and attend to piton area $F_h = \frac{\pi}{4} Dh^2$. thoroughface area of valve clearance

$$F_f = 2\pi N \beta h \Sigma Di$$

so

$$V_f = \frac{\mu \lambda v Dh^2}{8 N\beta h (1 - \frac{\theta}{180}) \Sigma Di} \qquad \ldots(12)$$

replace into (6) with 7 (12), may gect peak frequency of the valve jet noise

$$f_s = \frac{s \cdot \mu \cdot \lambda v \cdot Dh^2}{8 Nh\beta (1 - \frac{\theta}{180}) \Sigma Di \cdot \sqrt{h D_1}}$$

[5] material point out when jet diameter is bigger, Strouhal's number upper limit s=0.2 but equivalent jet diameter of air compressor valve clearance all are less, so s=0.15

$$f_s = \frac{6.63 \mu \lambda v Dh^2}{N\beta h \sqrt{h D1} (1 - \frac{\theta}{180})} \times 10^{-3} \text{ (Hz)} \qquad \ldots(13)$$

in the formula μ, D_h, N, h, D1, ΣDi, are know number of air compressor, jest λ_v, θ, β are indirect know number, they still need compute or consult curve, compute λv, θ, by (10) (11) in the compution relate to relative clearance α, clearance expand index m may consult below:

large mediun-sized air compressor:

exhaust pressure $\leq 20 kgf/cm^2$ $\alpha = 0.07-0.12$

$> 20-231 kgf/cm^2$ $\alpha = 0.12-0.16$

small sized exhaust measure $< 0.2 m^3/min$ $\alpha = 0.088-0.10$

$> 0.3 m^3/min$ $\alpha = 0.035-0.05$

air compressor speed $n \leq 200$ r.p.m; $m=1.2-1.3$

$200 \leq n \leq 500$ r.p.m; $m=1.25-1.35$

$n > 500$ r.p.m; $m=1.4$

coefficient β of valve can be consulted from the curve which was offered by M.E.Fleker[6]

so, peak frequency of the pulsating airflow noise and peak frequency of the valve jet noise have been made, their repeated addition obtain peak frequency of air compressor inlet

$$f = \begin{cases} f_m = \frac{kn}{60} \text{ (single acting } n > 1200, \text{double acting } n > 600) & (14a) \\ f_s = \frac{6.63 \mu \lambda v \cdot D_h^2}{N \beta h \sqrt{hD1} (1 - \frac{\theta}{180}) \Sigma Di} \times 10^{-3} & \ldots(14b) \end{cases}$$

Below, interpret domain of (14a). From the above analysis we can known that peak frequency should have two, one of them is f_m, the other is fs. On its frequency spectrum curve there should be also two, but in effect there isn't. Some compressors only have f_s haven't f_m, this is people's zone of audible frequency concem because people's audible frequency is 20-20000Hz, higher than 20000Hz cann't be listened by people. If sound is made by compressor in the two scopes people cann't listen, so the sound isn't noise. This will not research in the command domain of noise and use not to express it with actave frequency. If replace into (14a) with critical sound frequency f=20Hz, which people ~~people~~ can listen, it's possible to get airflow pulsation noise crical speed n_{min}, $n_{min}=\frac{60\times20}{K}=\frac{1200}{K}$

As for single acting compressor, because of K=1 it's possible to produce airflow pulsation noise critical speed which is 1200r.p.m; double acting compressor K=2, it's possible to get airflow pulsation noise critical speed which is 600r.p.m. so rotational speed low 1200r.p.m single acting compressor and rotational speed low 600r.p.m. double acting compressor will not get airflow pulsation noise, on their suction noise frequency spectrum curve only have one peak, other compressors, their suction noise have two peak frequencies, on their suction noise frequency spectrum curve have two peak, so (14a)'s field of definitaions is single: n >1200r.p.m, double: n >600r.p.m

AN REAL EXAMPLE FOR THE RECKONING OF PEAK NOISE FREQUENCY OF AIR COMPRESSOR INLET

Below take for example a square air compressor type L3.5-20/7
known n=980r.p.m u=3.92m/s D_h=0.38 h=0.0019 D1=0.1975m N=2
ΣD_i=0.81 m b=0.005 ε_1= 3 α=0.15 K=2 m=1.4
count:

1) judge peak frequency number

because L3.5-20/7 is double acting compressor, it's speed 980r.p.m, higher than critical speed 600r.p.m, so peak frequency have two, f_m and f_s.

2) count f_m by (14a)

$$f_m = \frac{20 \times 980}{60} = 32.7$$

3) count f_s

first count volume coefficient λ_v by (10)

$$\lambda v = 1- 0.15 \left(3^{\frac{1}{1.4}} - 1 \right) = 0.82$$

second count θ by (11), valve closing angle degree is θ

$$\theta = \cos^{-1} (0.82 \times 2-1) = 50$$

because $\frac{h}{b}$ = 0.38

Seek out the valve flow coefficient β = 0.5 by Fig.4 and substitute the relative number into (14b)

$$f_s = \frac{6.63 \times 3.92 \times 0.82 \times 0.38^2}{2 \times 0.5 \times 0.0019 \sqrt{0.0019 \times 0.1975}\,(1-\frac{50}{180})} \times 10^{-3} = 143$$

so L3.5-20/7 air compressor suction noise peak frequency have two: 32.7 and 143 (Hz), they are 31.5 and 125 (Hz) in octave frequency, this is agreeable to ractically surveying Fig.1.

RELIABILITY OF THE RECKONING FORMULA FOR PEAK FREQUENCY PROVING

Here is the inlet noise frequency of eight different air conpressors whose exhaustion capacity is o.6-40m^3/min. and take it as example for examining the reliablety of this

1) Number verifing of inlet noise peak frequency

With rameters of eight kinds air compressor make List 1. Contrast reckoning and measuring of the eight kinds air compressor inlet noise frequency numbers in the List 2. From domain (14a) known, when single acting compressor speed < 1200 r.p.m; double acting compressor speed <600r.p.m will not get airflow pulsation noise, compressor that all speeds are in this extent, their noise of inlet only one peak frequency. Other compressors have two peaks frequency.

From List 2 we can see: two numbers of both agree well. This is why 2Z-3/8-1, V-3/8-1, 3L-10/8 air compressors inlets frequencies spectrum only have one peak, but 2V-0.6/7, L2-10/8-1, L3.5-20/7, L3.3I-30/8 have two peaks.

2) Verifing of inlet noise peak octave frequency

Reckoning and measure numbers of inlet noise peak octave frequency of the eight kinds of air compressors, both agree well.

So reckoning formula is correct and convenient.

CONCLUSION

1. Noise frequency spectrum of air compressor inlet is broad band and continuance frequency spectrum, it's peak must get in low frequency. Number peak is determined by the speed and the acting method (single acting and double acting).

2. Noise peak frequency of air compressor inlet is determined by noise peak frequency of airflow panting and noise peak frequency of suction valve jet.

3. Amplitude value in sound-power frequency spectrum of inlet noise air compressor, since f_s to rear basically in general is function of Strouhal's number.

Amplitude value of f_s front is determined by airflow and Strouhal's number.

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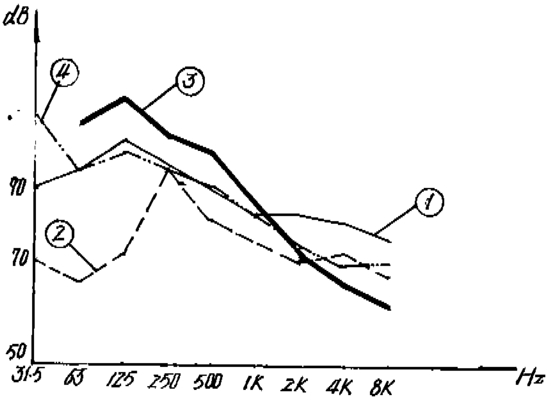
List 1

order number	type	air displacement (m^3/min)	compress series	single double acting	speed r.p.m	piston speed u(m/s)	the first grade bore Dh (mm)	the first grade compresser propertion ε_1	suction valve number
1	2V-0.6/7	0.6	1	single	1450	2.66	90	8	1
2	2Z-3/8-1	3	2	..	730	2.92	250	3	1
3	V-3/8-1	3	2	..	980	3.59	210	3	2
4	L2-10/8-1	10	2	double	980	3.92	275	3	2
5	3L-10/8	10	2	..	480	3.2	300	3	2
6	L3.5-20/7	20	2	..	980	3.92	380	3	2
7	L3.5I-20/7	20	2	..	980	3.92	380	3	2
8	L5.5-40/8	40	2	..	600	3.6	560	3	3

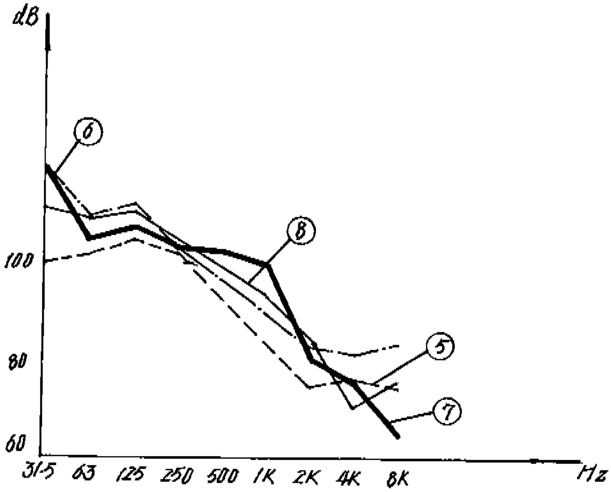
		relatively clearence α(0/0)	expansion index m	valve passage b(mm)	valve-flake rising-distance h (mm)	pre annulation valve-plake's medium-diamater's sum ΣD_i (mm)	the most exterior annulation valve-flake's medium-diamater D1 (mm)
		5	1.4	5	2.6	42.5	42.5
		12	1.4	7.5	2.5	380	140
		12	1.4	7.5	2.5	252	120
		14	1.4	5	1.7	440	147.5
		12	1.35	6.5	2.7	354	130.5
		15	1.4	5	1.9	810	197.5
		15	1.4	5	1.8	720	182.5
		15	1.4	5	2.4	810	197.5

List 2

order number	type	$\frac{h}{b}$	β	λ_v $\lambda_v=1-\alpha(\varepsilon_1^{\frac{1}{m}}-1)$	θ $\theta=\cos^{-1}(2\lambda_v-1)$ angle	number of suction noise peak frequency: reckoning	real	reckoning peak frequency of suction noise: reckoning f_n (Hz)	real f_s (Hz)	peak frequency of octave: reckoning	real
1	2V −0.6/7	0.52	0.46	0.83	48.8	2	2	24.2	304	31.5 , 250	31.5 , 250
2	2Z −3/8 −1	0.33	0.53	0.86	44.4	1	1	—	147	125	125
3	V −3/8 −1	0.33	0.53	0.86	44.4	1	1		104	125	125
4	L2 −10/8 −I	0.34	0.52	0.82	50	2	2	32.7	181	31.5 , 250	31.5 , 250
5	3L −10/8	0.42	0.48	0.85	45.7	1	1		126	125	125
6	L3.5 −20/7	0.38	0.50	0.82	50	2	2	32.7	143	31.5 , 125	31.5 , 125
7	L3.51 −20/8	0.36	0.52	0.82	50			32.7	174	31.5 . 125	31.5 , 125
8	L5.5 −40/8	0.48	0.47	0.82	50	2	2	20	142	31.5 , 125	31.5 , 125



(a)



(b)

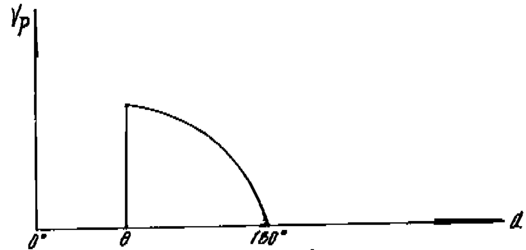
Fig. 1

1. ZY - 0.6/7 2. 2Z - 3/8 - 1

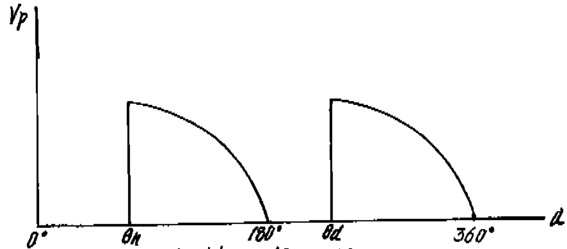
3. Y - 3/8 - 1 4. LZ - 10/8 - I

5. 3L - 10/8 6. L3.5 - 20/7

7. L3.5I - 20/8 8. L5.5 - 40/8



(a) single acting Vp



(b) double acting Vp

Fig 2

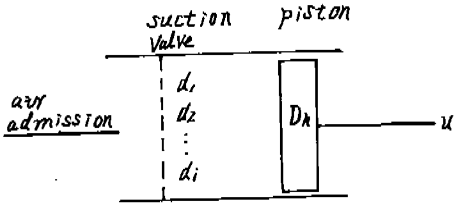


Fig 3
Suction Valve
equivalency

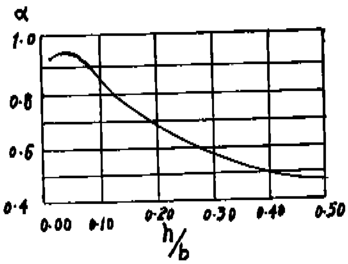


Fig 4