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

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ABSTRACT

The peak noise frequency of air compressor inlet is an important parameter when designing suction muffler of air compressor. From the aerodynamics of flow 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 has verified the reliability of the reckoning formula with spectrum of eight different air compressors, the result of noise capacity of which is 6.4~40m³/min and the calculation agrees well with the result. The formula will render some convenience for designing suction muffler of air compressor.

SYMBOLS

C      piston instantaneous velocity         m/s
Dh     piston diameter                       m
Fg     thoroughface area of suction pipe     m²
Fh     piston area                           m²
Fr     thoroughface of valve clearance       m²
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
Vr     airflow velocity of valve clearance    m/s
Vg     airflow velocity of suction pipe       m/s
d     equivalent jet diameter                 m/s
fm     peak frequency of pulsating airflow noise Hz
peak frequency of valve jet noise \( f_p \) Hz

diameter of valve flake rising distance \( h \) m

clearance expansion index \( m \) r.p.m.

air compressor speed \( n \) m/s

radius of crank throw \( r \)

piston mean velocity \( u \)

degree

the first grade relatively clearance volume of the rotative angle of crankshaft \( \alpha \)

suction valve closing angle of the first grade \( \theta_s \)

suction valve closing angle of crankshaft-side \( \theta_h \)

suction valve closing angle of lid-side \( \theta_i \)

the first grade compressor proportion \( \lambda \)

connecting rod proportion \( \lambda_v \)

volume coefficient of the first grade \( \lambda_v \) 0/0

compressor angular velocity \( \omega \)

suction valve flow coefficient of the first grade \( \beta \)

**INTRODUCTION**

It is known to all that the noise of air compressor inlet is the 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 foremost research problem of air compressor noise control. When we begin to design a suction 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, L3.5-10/8, L3.5-20/7, L3.5-20/8, 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 similar paper is biffish. The derivative basis of this paper is the valve jet aero-thermo-acoustics. To use the derivative reckoning formula is very easy and there is no need for computer simulation.

DERIVATION OF THE RECKON FORMULA

Noise of the air compressor inlet is the piling up of every noise which can be produced from the air compressor inlet to suction system containing 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, friction 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 friction noise should be negligible.

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 aeroacoustics 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:

\[
V = \frac{Fc \cdot C}{F_p}
\]

\[
C = \varphi \cdot \omega \cdot \left( \sin \alpha + \frac{1}{2} \sin 2\alpha \right)
\]

so

\[
V = \frac{\varphi \cdot \omega \cdot Fc}{F_p} \left( \sin \alpha + \frac{1}{2} \sin 2\alpha \right) \quad \ldots (1)
\]

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 \( \theta_0 \) or \( \theta_d \), so the functional equation of the airflow velocity of the suction pipe is single acting.

\[
V = \begin{cases} 
0 & 0 \leq \alpha \leq \theta_0 \\
\frac{\varphi \cdot \omega \cdot Fc}{F_p} \left( \sin \alpha + \frac{1}{2} \sin 2\alpha \right) & \theta_0 < \alpha < \theta_d \\
0 & 180^\circ < \alpha < 360^\circ 
\end{cases} \quad \ldots (2)
\]

double acting

\[
V = \begin{cases} 
0 & 0 \leq \alpha \leq \theta_0 \\
\frac{\varphi \cdot \omega \cdot Fc}{F_p} \left( \sin \alpha + \frac{1}{2} \sin 2\alpha \right) & \theta_0 < \alpha < 180^\circ \\
\frac{\varphi \cdot \omega \cdot Fc}{F_p} \left( \sin \alpha + \frac{1}{2} \sin 2\alpha \right) & 180^\circ < \alpha < \theta_d \\
0 & \theta_d < \alpha < 360^\circ 
\end{cases} \quad \ldots (3)
\]

Based on the formula (2), (3) we see the velocity \( V_p \) isn't elementary 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 transform them into Fuller's functions with ease in looking for the biggest harmonic. This harmonic functional frequency is the peak frequency of the airflow velocity \( V_p \) of the air compressor suction pipe.

\[
v_p = \frac{1}{2} \sum_{n=1}^{\infty} \left( a_n \cos n\alpha + b_n \sin n\alpha \right) \quad \ldots (4)
\]

\[
a_n = \frac{\varphi \cdot \omega \cdot Fc}{\pi} \int_0^{2\pi} \left| \frac{V_p}{p} \right| \cos n\alpha d\alpha \quad (n=0,1,2,\ldots)
\]

\[
b_n = \frac{\varphi \cdot \omega \cdot Fc}{\pi} \int_0^{2\pi} \left| \frac{V_p}{p} \right| \sin n\alpha d\alpha \quad (n=1,2,\ldots)
\]

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

So the peak frequency of the single acting compressor airflow velocity \( V_p \) is frequency of the compressor rotational speed. And the peak frequency of the double acting compressor airflow velocity \( V_p \) is twice as the frequency of the compressor rotational

\[
\text{means: } f_m = \frac{k \cdot n}{60} \quad \cdots (5)
\]

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 components are rich, and certain frequency, there is a higher peak: it's spectrum of sound capacity is the function of Strouhal's number \( f_d \) \( \cdots (3) \); it's peak sound spectrum will also change when jet velocity changes, but the peak frequency of this spectrum is always on the specifical frequency \( f \), whether it changes.

\[
f = \frac{sV}{d} \quad \cdots (6)
\]

\( s=0.15\text{--}0.20 \)

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

\[
\begin{align*}
\text{air} & \quad \text{Dh} \\
\text{admission} & \quad u \\
\text{di, d2, ..., d4: per valve clearance equivalent diameter}
\end{align*}
\]

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 peak frequency is different. If
So, we can see valve as a phonation-eventment, which produce many frequencies tones at the same time. If airflow pass several holes at the same time, it can produce several frequencies tone. Low or high of valve clearance jet sound-power is decided by every valve clearance valve, from Lighthill's law know, in supplies condition, jet noise is directly proportional to jet diameter square. For the valve in working order, the most exterior annular 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 annular valve clearance valve-flake is $D_1$, valve-flake rising-distance is $h$, according to area conversion written it's jet equivalent diameter $d = \sqrt{hD_1}$ ...(7)

Airflow speed of valve clearance $V_f$ through (1) formula may look for, exchange of $F_f$ for $F_f$ in the (1) formula, as it is, crank up then angle in $\theta - 180^\circ$the suction valves, so airflow average speed of valve clearance $V_f$: 

$$V_f = \frac{V_f}{\frac{180}{\pi}} \frac{1}{(180-\theta)} \int \frac{\sin \alpha + \frac{\alpha}{2} \sin 2\alpha}{2} d\alpha$$

$$\omega = \frac{\pi n}{30}$$

and piston velocity $U = \frac{\pi n}{15}$

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

$$V_f = \frac{90 \cdot U \cdot F_n}{(180-\theta) F_f} \left[ (1-\cos \theta) - \frac{\alpha}{4} (1 - \cos 2\theta) \right]$$

Usually, the first grade compress proportion $\xi_1$ is taken small, it's only $0.9 - 0.95$ multiple of equal compress proportion designer in order to lessen measure and weight of compressor, in order to heigher volume of coefficient $\lambda_V$ must restrict in the least clearance volume of the first grade, cause the first grade suction valve is closing early, mean $\theta$ is less, so the second item $\lambda (1-\cos 2\alpha)$ in bracket of aforesaid formula is less than item infront, may neglect. So aforesaid formula simplify

$$V_f = \frac{90 \cdot U \cdot F_n}{(180-\theta) F_f} (1 + \cos \theta)$$

because closing angle of suction valve is:

$$\theta = \cos^{-1} \left[ 1 - 2\alpha (\xi_1 \frac{1}{2} - 1) \right]$$

$$\lambda_V = 1 - \alpha (\xi_1^2 - 1)$$

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

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

replace into (9) with (11) and attend to piton area $F = \pi D_h^2 / 4$ thoroughface area of valve clearance

$$P_f = 2\pi N \beta h \Xi D_i$$

so

$$v_f = \frac{\pi \lambda v D_h^2}{3 \pi h (1 - \theta) \Xi D_i} \quad \ldots(12)$$

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

$$f_s = \frac{s \cdot U \cdot \lambda v \cdot D_h^2}{8 \pi N h (1 - \theta) \Xi D_i \cdot \sqrt{h D_i}} \quad \ldots(13)$$

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 $s = 0.15$

$$f_s = \frac{6.63 \cdot U \cdot \lambda v \cdot D_h^2}{N \beta h \sqrt{h D_i} (1 - \theta)} \times 10^{-3} \text{ (Hz)} \quad \ldots(13)$$

in the formula $U$, $D_h$, $N$, $h$, $D_i$, $\Xi D_i$, are know number of air compressor, jet $\lambda v$, $\theta$, $\beta$ are indirect know number, they still need compute or consult curve, compute $\lambda v$, $\theta$, by (10) (11) in the computation relate to relative clearance $\alpha$, clearance expand index $m$ may consult below:

large medium-sized air compressor:

- exhaust pressure $\leq 20 \text{ kgf/cm}^2$
- $> 20 - 231 \text{ kgf/cm}^2$ $\alpha = 0.07 - 0.12$
- $> 0.12 - 0.16$

small sized exhaust measure $< 0.2 \text{ m}^3/\text{min}$ $\alpha = 0.088 - 0.10$
- $> 0.3 \text{ m}^3/\text{min}$ $\alpha = 0.035 - 0.05$

air compressor speed $n \leq 200 \text{ r.p.m.; } m = 1.2 - 1.3$
- $> 200 - 500 \text{ r.p.m.; } m = 1.25 - 1.35$
- $n > 500 \text{ r.p.m.; } m = 1.4$

coefficient $\beta$ of valve can be consulted from the curve which was offered by H.E. Fleker

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

$$f = \begin{cases} f_m \frac{\text{kn}}{30} & \text{(single acting } n \geq 1200, \text{ double acting } n \geq 600) \quad (14a) \\ f_s = \frac{6.63 U \lambda v \cdot \frac{D_h^2}{h}}{N \beta h \sqrt{h D_i} (1 - \theta) \Xi D_i} \times 10^{-3} \quad \ldots(14b) \end{cases}$$

Below, interpret domain of (14a). From the above analysis we
can known peak frequency should have two, one of them is $f_m$, the other is $f_s$. On the frequency spectrum curve there should be also two, but in effect there isn't. Some compressors only have $f_m$ haven't $f_s$, this is people's zone of audible frequency concern because people's audible frequency is 20-20000Hz, higher than 20000Hz can't be listened by people. If sound is made by compressor in the two scopes people can'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 active frequency. If replace into (14a) with critical sound frequency $f=20Hz$, which people can listen, it's possible to get airflow pulsation noise critical speed $n_{min}$ $\frac{n_{min}}{K} = \frac{20 \times 980}{60} = 3200$

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 definitions 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 $n=980r.p.m \; u=3.92m/s \; D_h=0.38 \; h=0.0019 \; D_1=0.1975m \; N_m=2$

\[ \Sigma D_i=0.81 \; m \; b=0.005 \; \epsilon=0.15 \; \alpha=0.15 \; K=2 \; m=1.4 \]

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 $\lambda_v$ by (10)

\[ \lambda_v = 1- \frac{0.15}{3^{1.4} - 1} = 0.82 \]

second count by (11), valve closing angle degree is $\theta$

\[ \theta = \cos^{-1} \left( 0.82 \times 2-1 \right) = 50 \]

because $h = 0.38$

Seek out the valve flow coefficient $\beta = 0.5$ by Fig.4 and substitute the relative number into (14b)
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 agreeable to practically surveying Fig.1.

RELIABILITY OF THE RECKONING FORMULA FOR PEAK
FREQUENCY PROVING

Here is the inlet noise frequency of eight different
air compressors whose exhaust capacity is 0.6-40 m3/min. and
take it as example for examining the reliability of this
1) Number verifying of inlet noise peak frequency

With rameters of eight kinds air compressor make list 1.
Contrast reckoning and measuring of the eight kinds air com­
pressor 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 < 600 r.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 22-3/8-1, 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

<table>
<thead>
<tr>
<th>Order number</th>
<th>Type</th>
<th>Air displacement ( m^3/min )</th>
<th>Compress series</th>
<th>Single double acting</th>
<th>Speed ( rpm )</th>
<th>Piston speed ( m/s )</th>
<th>The first grade bore ( D_h ) (mm)</th>
<th>The first grade compressor proportion ( c_1 )</th>
<th>Susteen valve number</th>
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<tbody>
<tr>
<td>1</td>
<td>2y-0·6/7</td>
<td>0·6</td>
<td>1</td>
<td>Single</td>
<td>1450</td>
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<td>90</td>
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<td>2Z-3/8-1</td>
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<td></td>
<td>730</td>
<td>2·92</td>
<td>250</td>
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<td>3</td>
<td>V-3/8-1</td>
<td>3</td>
<td>2</td>
<td></td>
<td>980</td>
<td>3·59</td>
<td>210</td>
<td>3</td>
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<td>275</td>
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<td>300</td>
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<td>980</td>
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<td>560</td>
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<table>
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<tr>
<th>Relatively clearance ( \alpha (0/0) )</th>
<th>Expansion index ( a )</th>
<th>Valve passage ( b ) (mm)</th>
<th>Valve-flake rising-distance ( h ) (mm)</th>
<th>Pre annulation valve-flake’s medium-diameter’s sum ( \Sigma D_i ) (mm)</th>
<th>The most exterior annulation valve-flake’s medium-diameter ( D_1 ) (mm)</th>
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<tr>
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<td>1.4</td>
<td>5</td>
<td>2.4</td>
<td>810</td>
<td>197.5</td>
</tr>
<tr>
<td>Order number</td>
<td>Type</td>
<td>( \frac{h}{b} )</td>
<td>( \beta )</td>
<td>( \lambda' ) ( \lambda'_{w} = 1 - \alpha \left( \frac{L'}{L''} - 1 \right) )</td>
<td>( \theta = \cos^{-1} \left( 2 \lambda'_{w} - 1 \right) )</td>
</tr>
<tr>
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<td>-----------</td>
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<td>-------------</td>
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<tr>
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<td>2V - 0.6/7</td>
<td>0.32</td>
<td>0.46</td>
<td>0.83</td>
<td>48.8</td>
</tr>
<tr>
<td>2</td>
<td>2V - 3/8 - L</td>
<td>0.33</td>
<td>0.53</td>
<td>0.86</td>
<td>44.4</td>
</tr>
<tr>
<td>3</td>
<td>Y - 3/8 - 1</td>
<td>0.33</td>
<td>0.53</td>
<td>0.86</td>
<td>44.4</td>
</tr>
<tr>
<td>4</td>
<td>L2 - 10/8 - 1</td>
<td>0.34</td>
<td>0.52</td>
<td>0.82</td>
<td>50.0</td>
</tr>
<tr>
<td>5</td>
<td>3L - 10/8</td>
<td>0.42</td>
<td>0.48</td>
<td>0.85</td>
<td>45.7</td>
</tr>
<tr>
<td>6</td>
<td>L35 - 20/7</td>
<td>0.38</td>
<td>0.50</td>
<td>0.82</td>
<td>50.0</td>
</tr>
<tr>
<td>7</td>
<td>L351 - 20/8</td>
<td>0.36</td>
<td>0.52</td>
<td>0.82</td>
<td>50.0</td>
</tr>
<tr>
<td>8</td>
<td>L55 - 40/8</td>
<td>0.48</td>
<td>0.47</td>
<td>0.82</td>
<td>50.0</td>
</tr>
</tbody>
</table>

**Peak frequency of active reed:**

- \( 31.5, 250 \)
- \( 125 \)
- \( 125 \)
- \( 125, 250 \)
- \( 125 \)
- \( 125, 250 \)
- \( 31.5, 125 \)
- \( 31.5, 125 \)
- \( 31.5, 125 \)
- \( 31.5, 125 \)
Fig. 1

1. 2V - 0.6/1
2. 22 - 3/8 - 1
3. V - 3/8 - 1
4. L2 - 10/8 - 1
5. J4 - 10/8
6. L3-5 - 20/7
7. L551 - 20/8
8. L5-5 - 40/8
Fig 2

Fig 3
Suction Valve equivalency

Fig 4