1984

Decreasing the M Value of the Valves to Reduce the Air Dynamic Noise in the Compressor

X. Qian

Follow this and additional works at: http://docs.lib.purdue.edu/icec

http://docs.lib.purdue.edu/icec/485

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.
Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html
DECREASING THE M VALUE OF THE VALVES TO REDUCE THE AIR DYNAMIC NOISE IN THE COMPRESSOR

Qian Xinghua, Engineer, Xian Compressor Works, Xian, Shaanxi, The People’s Republic of China

ABSTRACT

Based on the low of the eighth power of speed of M.J. Lighthill this article has derived the calculating formula for the air dynamic noise power in the compressor, and analysed the factors to affect the noise, also indicated that the dynamic noise in the compressor mainly comes from the valve jet noise; the principle to reduce the dynamic noise is to get a small Reynolds number \( R_e \): the emphasis to reduce the noise is on the jet turbulent flow noise; the efficient way to reduce the noise is to decrease the valve Mach number.

INTRODUCTION

The noise from the jet noise is named air dynamic noise, which is the main source of the noise in the compressor. To prevent noise pollution, it is necessary to control it. This problem is an important subject to research, and some accomplishment on mufflers and acoustical enclosures have been reached. This article provides a method to reduce the air dynamic noise in the compressor by decreasing the valve Mach number. It also says that the basic way to reduce the noise is to reduce the noise power in the noise source. The muffler and acoustical enclosure are only the assistant way.

From acoustics, turbulent flow comes from throttling inherently, and the flow passage consists of a few serial throttlings. So there comes the air dynamic noise at the throttlings. How to reduce the air dynamic noise in the compressor from its root? The article says that the principle to reduce the air dynamic noise is to get a small Reynolds number \( R_e \) and the emphasis is on the valve jet turbulent noise; The efficient way to reduce the noise is to decrease the valve Mach number.

THE PRINCIPLE OF A SMALL \( R_e \)

The source of air dynamic noise is at turb-
The throttling jet is an isentropic process, so
\[ p \rho^e = \text{constant} \]
Differentiate this and (2)
\[ \frac{dp}{p} = K \frac{d\rho}{\rho} \]
\[ \frac{d\rho}{\rho} = \frac{dp}{p} - \frac{dT}{T} \]
From those two
\[ \frac{dT}{T} = \frac{K-1}{K} \frac{dp}{p} \]
For the throttling in the compressor, the relative pressure loss is very small, so, the relative temperature drop is very small, too. To simplify the question the pressure and temperature before and after throttling are thought unchanged, \( p_g = p', T_g = T \). So from equations (2) (3) it is also thought that the density and sound speed are unchanged, \( \rho = \rho', c = c' \). Substitute the two and (4) to (1) and consider
\[ v = MC \]
The following formula can be drawn
\[ W = nKgPuD'M' \]
(6)
From (7), the factors to affect the throttling jet turbulent noise in the compressor may be found out:
1. The noise power \( W \) is directly proportional to the seventh power of the Mach number.
2. The noise power \( W \) is directly proportional to \( K \). So with the same noise level for a compressor with the medium of small \( K \), its Mach number may be chosen properly greater.
3. The noise power is directly proportional to the second power of \( D \), the cylinder bore and to \( u \), the piston speed. Also \( Q = \pi D^2 uA/4 \), therefore the noise power is directly proportional to \( Q \), so the noise power in a large compressor is greater than that of a small one.
4. The noise power is directly proportional to \( P \), jet pressure. So it can be concluded that to compare the noise of compressors the identical ambient is necessary, otherwise there is no comparison between the two. Take the air compressor as an example. The noise measured in plateau would be smaller. From the international pressure-height above sea level formula
\[ P = 1.033 \left(1 - 0.02257H\right)^{5.256} \]
\( H \): height above sea level, unit KM and differentiate (7)
\[ \frac{dW}{DH} = \frac{dW}{dp} \frac{dp}{DH} = \frac{-0.119W}{1 - 0.02257H} \]
Here \( H \) is of several kilometers, so
\[ \frac{dW}{W} = -0.0019H \]
It can be seen that the throttling jet noise power is inversely proportional to the height above sea level. If the height increases by \( 1/M \), the noise power will reduce by 0.1%. 5. The noise power is directly proportional to jet pressure \( P \), so among the valves in the same stage the discharge jet turbulent flow noise is greater than the suction.
6. Among the valves with the same name, the jet turbulent flow noise from the high pressure stage is greater than that from the low pressure stage. To explain this point by a compound compressor. The following formula can be drawn from (7)
\[ \frac{W_2}{W_1} = \frac{nKgP_2uD_2'M_2'p_2}{nKgP_1uD_1'M_1'p_1} = \frac{P_1}{P_2} \left( \frac{D_1}{D_2} \right)^{\left(\frac{M_1}{M_2}\right)^2} \]
From \( Q = \pi D^2 uA/4 \) and the ideal gas is supposed, the equation of state is
\[ PQ = \frac{FP}{RT} \]
therefore
\[ \frac{P_1}{P_2} = \frac{\pi D_1^2 u/4}{\pi D_2^2 u/4} = \frac{D_2^2}{D_1^2} \]
The compression ratio for one stage is
\[ \frac{P_2}{P_1} = \frac{\pi D_2^2 u/4}{\pi D_1^2 u/4} = \frac{D_1^2}{D_2^2} \]
Considering \( M_2 = M_1 \) generally
\[ \frac{W_2}{W_1} = \frac{\lambda_2}{\lambda_1} \frac{T_2}{T_1} \]
Generally speaking, the volumetric efficiency \( \lambda_2 \) at the second stage is less than \( \lambda_1 \) at the first stage and the suction temperature \( T_2 \) at the second stage is greater than \( T_1 \) at the first stage. So
\[ \frac{W_2}{W_1} > 1 \]
That is the noise from the high stage is greater than that from the low stage.

THE CALCULATING FOR AIR DYNAMIC NOISE POWER

The air dynamic noise is produced by accumulating each jet noise in the compressor. So the noise power equals the sum of all jet noise powers.

Firstly deriving the air dynamic noise power of the suction system. A serial throttling piping (shown in Fig.1) is used to be the equivalent to the serial throttling system.
Here $d_1$, $d_2$, ..., $d_i$ denote the diameters of all throttlings. $M_1$, $M_2$, ..., $M_i$ is each throttling jet noise powers. $D$ piston diameter $u$ piston speed

The following conclusions can be drawn:
1. The equivalent throttling diameter $d_e$ of the serial throttling system is smaller than any of the real throttling diameter $d_i$;
2. If the smallest one $d_{min}$ is much smaller than others, i.e. $d_{min} \ll d_i$, then it is thought $d_e = d_{min}$.

Among the throttlings the throttling diameter $d_v$ of the valve is much smaller than the others, that is $d_v < d_i$, so $d_e = d_v$. $N_v$ is the Mach number of the valve, then the equivalent Mach number would be $N_e = N_v$. Substitute them into (11)

$$W = n K g P u D^2 N_v^7$$

$N_v$ being the Mach number of the suction valve and according to (13), air dynamic noise power of the suction system can be expressed blow:

$$W = n K g P u D^2 N_v^7$$

That is the air dynamic noise power of the suction system depends on only the valve jet turbulent flow noise power, the others may be omitted.

For the discharge system,

$$W_d = n K g P u D^2 N_v d$$

So the conclusions above apply to the discharge system, too. In that case, the whole air dynamic noise power equals the sum of the jet turbulent flow noise powers of the suction valve and discharge valve, the other throttling noise may be omitted:

$$W = W_s + W_d$$

$$W = n K g P u D^2 N_v^7$$

For a compound compressor the air dynamic noise power

$$W = n K g P u D^2 N_v^7$$

$$W = n K g P u D^2 N_v^7$$

$$W = n K g P u D^2 N_v^7$$

Considering $E_1 = \frac{P_1}{P_s}$, $E_2 = \frac{P_2}{P_s}$ and usually $P_{S2} = P_{d1}$, $N_v = N_v s1 = N_v d1 = N_v s2 = N_v d2$, substitute them into the above equation and use equation (6)

$$W = n K g P_{S1} D^2 N_v^7$$

For the same $N_v$ and $E_1$, $E_2$, the air dynamic noise power is directly proportional to $P_{S1}$ and to the second power of cylinder bore $D^2$. Additionally it is also affected by the efficiency of inter cooler. When volumetric efficiency of the second stage is small and the suction temperature is higher, i.e. $\lambda_2 < \lambda_1$, $T_{S2} > T_{S1}$ then $\lambda_2 > \lambda_1$, $\frac{T_{S2}}{T_{S1}} > 1$, that would cause the noise to increase. For an ideal compressor because $T_{S1} = T_{S2}$,

$$\lambda_1 = \lambda_2, \quad E_1 = E_2 = 0$$

therefore the equation (15) becomes
For an ideal multiple (I) compressors, the air dynamic noise power can be calculated by
\[ W = 2n K g u P s 1 D_s^2 N_v^7 (1 + \varepsilon^4) \] (16)

According to equation (17), it is easy to estimate the air dynamic noise power. That is the noise power wouldn't be smaller than the ideal value which comes from the equation (17).

DECREASING THE MACH NUMBER OF THE VALVE TO REDUCE THE AERODYNAMIC NOISE POWER

Based on the analysis above it can be known that the air dynamic noise power is directly proportional to the seventh power of valve Mach number. So the most efficient way to reduce the noise is to decrease the valve Mach number. Differentiate(13) or (16), (17)

\[ \frac{dW}{W} = 7 \frac{dM_v}{M_v} \]

It can be expressed with the relative increment form

\[ \frac{dW}{W} = 7 \frac{\Delta M_v}{M_v} \] (18)

It can be seen that the increment of the Mach number would cause 7 times over that of the air dynamic noise power. A decrement of the Mach number by 1% would cause decrement by 7%.

The affect to sound pressure level by the valve Mach number will be studied below.

\[ L_p = 20 \log \left( \frac{P_s}{P} \right) \] (19)

Here

\[ P = \frac{1}{\beta} \int \frac{w e C}{\beta \pi} \]

\[ \rho = 400 \text{ rayl} \] (acoustic impedance)

For a free sound field \( \beta = 4 \), half free sound field, \( \beta = 2 \). There are several valves in a compressor. Suppose that the sound pressures tested at the given points are \( P_1, P_2, ..., P_i \) and the distances from the tested points are \( r_1, r_2, ..., r_i \), then the whole sound pressure level

\[ L_p = 10 \log \frac{P_1^2 + P_2^2 + ... + P_i^2}{P_s^2} \] (20)

Substitute (20) into the above

\[ L_p = 10 \log \frac{C \sum C N_v M_v^7 (P_s^2)}{\beta \pi P_s^2} \]

That is the formula calculating the sound pressure level in the compressor without any noise decay device. Differentiate (21)

\[ dL_p = 30.4 \frac{dM_v}{M_v} \]

It can be seen that the decrement of the valve Mach number by 1%, the sound pressure level would be reduced by 0.34 dB.

REMARK

At present, when designing the valve, compressor designers in all countries follow the three points of the regulation:
1. Small loss of the valve energy;
2. Valves are used in safety and reliably with long life;
3. The machinability of the valve should be easy.

It is thought in this article that another point should be added in, that is, the noise power of the valve should be small. Only the things that satisfy with the four points are perfect.

The principle to reduce the air dynamic noise in the compressor is of a small Re. The emphasis to reduce the noise should be put on reducing the jet turbulent flow noise. The efficient way to reduce the noise is to decrease valve Mach number.

When using the sound power level to test the compressor noise, the formula calculating the sound power, recommended by this article is useful to assess sound decay devices (mufflers, sound barriers, acoustical enclosures and the like.).

Limited by the conditions the affect to the whole compressor by the valve Mach number has not been experimented. However, the experiment of the affect to the air dynamic noise of single stage suction system by the suction valve Mach number has been done, and tested three suction valves which have different Mach numbers. The result is in concord with equation (22) My experiments have proved that decreasing the valve Mach number is very efficient to the reduction of air dynamic noise. This method has been carried out by the author on the compressor 12-10/8-I and the effect is prominent.
Sound velocity before jet
Sound velocity after jet
Piston diameter
Nozzle diameter
Equivalent diameter
Valve equivalent diameter
Weight rate of flow
Gravity acceleration
Height above sea level (KM)
Adiabatic index (ratio of specific heat cp)

Sound pressure level
Mach number
Equivalent throttling Mach number
Valve Mach number
Discharge valve Mach number
Suction valve Mach number
Jet pressure, sound pressure, atmospheric pressure
Discharge pressure
Suction pressure
Basic pressure
Discharge volume
Gas constant
Reynolds number
Discharge of measuring noise
Temperature before jet
Temperature after jet
Discharge temperature
Suction temperature
Piston speed
Jet velocity
Equivalent throttling velocity
Whole pressure ratio
Pressure ratio
Volumetric efficiency
Coefficient of viscosity

REFERENCES