The Design of the Acoustic Isolator Used in Acoustic Telemetry While Drilling

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Abstract: The acoustic isolator is positioned between the bit and the transmitting transducer, it can attenuate a variety of front end of the drill noise signal to avoid the acoustic noise signal propagation along the drill collar, affecting the acoustic signal transmission and used to weaken the influence of downlink acoustic signal with the uplink channel produced by the transducer. This paper improved the characteristic analysis of tapered acoustic transmission model, designed of the size and structure of the acoustic isolator which has the transmission frequency channel of in 700~800Hz and 1200~1300Hz interval.

Keywords: Acoustic isolator, acoustic telemetry while drilling, acoustic transmission, modeling analysis.

1. INTRODUCTION

The wireless measurement while drilling system is widely used in the field of drilling, can real-time detect and upload the formation parameters and borehole attitude parameters in the drilling process, to ensure the well trajectory with the requirements of drilling. The transmission mode of MWD are mud pulse and electromagnetic waves, acoustic and optical fiber etc. The mud pulse and electromagnetic wave has widely used in oilfield. Affected by the drilling fluid, the mud pulse mwd cannot be used in air drilling, foam drilling condition, and the transmission capacity is limited, the domestic instrument’s rate can reach 3-5bps, the maximum rate can reach 10bps abroad. The method of electromagnetic waves MWD cannot be used in the low resistivity formation, and the transmission distance is limited, and not suitable for deep well operation [1-4].

The acoustic telemetry while drilling technique using elastic waves for data transmission in the drill column, cannot be influenced by drilling fluid and formation resistivity. With the influence of drill string’s cycle section and drilling noise, the theoretical rate of acoustic transmission can reach 100bps [1]. In 2007, XACT downhole telemetry inc. published the transmission rate of downhole acoustic transmission system to 20bps, and the test well depth is 2500m [5, 6].

Compared with the acoustic information transmission system used in production wells, the entire system is the first affected by the noise in the process of drilling and the wave reflected by the BHA below the telemetry system. This kind of noise is transmitted to the acoustic transmission channel, causes interference with the acoustic telemetry signal. The downhole BHA cannot be designed according to the characteristics of acoustic telemetry transmission system, therefore the acoustic isolator must be designed to eliminate acoustic reflection noise [1].

The acoustic telemetry instruments while drilling are mainly composed of acoustic transmitting transducer and receiving transducer, acoustic isolator and the supporting electronic circuit components. Acoustic signals generated by acoustic transmitting transducer propagates to the ground along the drill pipe, at the same time, the acoustic signal is downward transmitting, while the signal contact to the lower BHA and so on, the reflected signal cause serious interference on the uplink signal; the drilling noise also produce serious interference on the uplink signal.

The acoustic isolator body is positioned between the bit and the transmitting transducer, it has two major functions: attenuating the drilling noise through the acoustic isolator above the bit, avoid the acoustic signal propagating along the drill collar and affecting the acoustic signal telemetry; weakening the influence of acoustic reflected signal is produced by transmitting transducer for the uplink channel.

In the acoustic telemetry signal transmission process while drilling, the noise generated by the drill bit are mainly in the low frequency band below 600Hz, the transmitting band signal is chose between 700Hz and 2000Hz [7-10]. This paper mainly completes the design of acoustic isolator with transmission channel frequency in 700~800Hz and 1200~1300Hz interval.

2. THE CREATION OF A TRANSMISSION MODE

The calculation methods for acoustic transmission in the periodic drill pipe mainly include: Douglas finite difference method and the equivalent the acoustic transparent film method [11]. The former can only calculate the acoustic transmission characteristics of periodic drill pipe with two kinds of cross section, the latter can calculate the sound transmission characteristics of various cylindrical section
combined pipe structure, but for the tapered rod and various special-shaped column. The design of acoustic isolator body is mainly dependent on finite element software and the model machining measurement now, and there is no a practical fast estimate method [12-20].

In this paper, according to the design characteristics of acoustic isolator, using the longitudinal vibration theory, the four transfer matrix of terminal parameters were given which is about the vibration of different with variable cross section cylindrical drilling pipe or the gradual change section conical drilling pipe, the acoustic isolator of arbitrary cross section combined structure vibration transmission characteristics was got with certain boundary conditions, the input of acoustic isolator structure and transfer impedance was given out, and the relationships of the two ends’ vibration velocities and the input output force of acoustic isolator was given out, thus the engineering calculation method of modeling is completed about the small scale gradual change section acoustic isolator.

2.1. The Establishment of Various Tapered Cylindrical Pipe Vibration Equation and its Four End Parameter Equation

1. The vibration equation of variable cross section pipes

\[ \sigma = \left( \frac{\partial \xi}{\partial x} \right) \frac{dx}{dx} = \frac{\partial \xi}{\partial x} \]

- The longitudinal stress \( T \) is the compression force on the per unit area of the cross section

\[ T = \frac{F}{s(x)} \]

and \( s(x) \) is the cross-sectional area of the pipe at position \( x \).

Among them: \( c^2 = \frac{Y}{\rho} \), is the longitudinal wave velocity in the material.

Considering the simple harmonic vibration of the structure \( \xi = \xi_0 e^{j\omega t} \), Then can be got:

\[ \frac{\partial^2 \xi}{\partial x^2} + \frac{1}{s(x)} \frac{s(x)}{\partial x} \frac{\partial \xi}{\partial x} + k^2 \xi = 0 \quad (1) \]

Among them: \( k = \omega/c \) is the longitudinal wave number, this is the displacement equation of one dimensional longitudinal vibration of the variable section bar.

The velocity of the simple harmonic vibration

\[ v = \frac{\partial \xi}{\partial t} = j \omega \xi e^{j\omega t} \]

the vibration displacement in the pipe is the independent variable of axis and time, the equation (1) can be written in the form of differential equations of particle vibration velocity:

\[ \frac{\partial^2 v}{\partial x^2} + \frac{1}{s(x)} \frac{s(x)}{\partial x} \frac{\partial v}{\partial x} + k^2 v = 0 \quad (2) \]

The transfer force of the pipe is:

\[ F = Y_s(x) \frac{\partial \xi}{\partial x} = -j Y_s(x) \frac{\partial v}{\partial \omega} \]

2) Solving the equation is divided into two types: uniform section cylindrical pipe; 2. variable cross-section uniform cylindrical pipe.

(1) uniform section cylindrical pipe

As shown in Fig. (2), among them: \( z = \rho c s_0 \), the above four equations can be solved to get the relationship between input and output, and be written in the form of in four end parameter matrix:
\[
\begin{bmatrix}
v_1 \\
F_1
\end{bmatrix} =
\begin{bmatrix}
\cos kl & \frac{1}{jz} \sin kl \\
-jz \sin kl & \cos kl
\end{bmatrix}
\begin{bmatrix}
v_2 \\
F_2
\end{bmatrix}
\] (4)

The elements in the matrix above formula are four end parameters for uniform cylindrical pipe vibration. They describe the relationship between the input and output vibration acceleration of the ends of cylindrical pipe with elastic force.

(2) Variable cross-section uniform cylindrical pipe

If the pipe is a variable cross-section column, the cross-sectional area \( s(x) \) is a function of position coordinates, then solving the equations is more complex. The vibration equation of Variable cross-section column pipe is simplified, then give their four end parameter equation.

1. Linear tapered pipe

As shown in Fig. (1), the cross-sectional area of the taper pipe can be expressed as:

\[
s(x) = \pi r(x)^2 = \pi \left[ \frac{R_i - R_o}{l} x + R_i \right]^2
\] (5)

The relationship of input and output can be obtained by solving the above equation:

\[
v_1 = \alpha_1 v_2 + \alpha_2 F_2 \\
F_1 = \alpha_3 v_2 + \alpha_4 F_2
\] (6)

Among them:

\[
\alpha_1 = \frac{g_1 f_2 - g_2 f_1}{g_2 f_2 - g_1 f_1} \\
\alpha_2 = \frac{1}{jz_2} \frac{g_2 f_2 - g_1 f_2}{g_2 f_2 - g_1 f_1} \\
\alpha_3 = jz_1 \frac{f_2 g_1 - f_1 g_2}{f_2 g_2 - f_1 g_1} \\
\alpha_4 = \frac{z_1}{z_2} \frac{f_2 g_1 - g_2 f_1}{f_2 g_2 - g_1 f_1}
\]

The equation of the tapered pipe’s four terminal network parameters can be written in matrix form:

\[
\begin{bmatrix}
v_1 \\
F_1
\end{bmatrix} =
\begin{bmatrix}
\alpha_{11} & \alpha_{12} \\
\alpha_{21} & \alpha_{22}
\end{bmatrix}
\begin{bmatrix}
v_2 \\
F_2
\end{bmatrix}
\] (7)

2. The equal wall thickness hollow tapered pipe

In ends of the tubing collar have a threaded equal wall thickness hollow tapered rod, as shown in Fig. (3). The solution can be obtained as the follows four terminal network parameter matrix can be obtained by solving the equation:

\[
\begin{bmatrix}
v_1 \\
F_1
\end{bmatrix} =
\begin{bmatrix}
\beta_{11} & \beta_{12} \\
\beta_{21} & \beta_{22}
\end{bmatrix}
\begin{bmatrix}
v_2 \\
F_2
\end{bmatrix}
\] (8)

Among them:

\[
\beta_{11} = \frac{g_2 f_1 - g_1 f_2}{f_2 g_2 - f_1 g_2} \\
\beta_{12} = \frac{1}{jz_2} \frac{f_2 g_1 - f_1 g_2}{f_2 g_2 - f_1 g_1} \\
\beta_{21} = jz_1 \frac{g_2 f_1 - g_1 f_2}{g_2 f_2 - g_1 f_1} \\
\beta_{22} = \frac{z_1}{z_2} \frac{g_2 f_1 - g_1 f_2}{g_2 f_2 - g_1 f_1}
\]

3. The hollow straight tapered pipe

As shown in Fig. (4), the expression of the solution is as same as the Linear straight tapered pipe, respectively making \( x = 0 \) and \( x = l \), the velocity and elastic force expression of the pipe’s ends is got. After simplification, the four end parameter matrix equation of the hollow straight tapered pipe is:

\[
\begin{bmatrix}
v_1 \\
F_1
\end{bmatrix} =
\begin{bmatrix}
\gamma_{11} & \gamma_{12} \\
\gamma_{21} & \gamma_{22}
\end{bmatrix}
\begin{bmatrix}
v_2 \\
F_2
\end{bmatrix}
\] (13)

Among them the four end parameters \( \gamma_{11}, \gamma_{12}, \gamma_{21}, \gamma_{22} \) are as same as the linear tapered pipes’ \( \alpha_{11}, \alpha_{12}, \alpha_{21}, \alpha_{22} \), just the height of the conical tip is changed from \( a \) to \( b \).
With tapered bore cylindrical pipe

\[ \begin{bmatrix} v_1 \\ F_1 \end{bmatrix} = \begin{bmatrix} x_{11} & x_{12} \\ x_{21} & x_{22} \end{bmatrix} \begin{bmatrix} v_2 \\ F_2 \end{bmatrix} \]  (14)

among them:

\[ x_{11} = \frac{-f_2}{g_2 (R-R_1) (f_1 \sin kl - f_2 \cos kl)} \]  (15)

\[ x_{12} = \frac{g_2 \sin kl}{g_1 (R-R_1)} \]  (16)

\[ x_{21} = \frac{jz_2}{k} \frac{R_D f_2 - k (R-R_1) f_1}{g_1 (R-R_1)^2} \]  (17)

\[ x_{22} = \frac{jz_2}{k} \frac{R_D \sin kl + k (R-R_1) \cos kl}{g_1 (R-R_1)^2} \]  (18)

\[ f_1 = g_1 \cos kl + g_2 k \sin kl \]  (19)

\[ f_2 = g_1 \sin kl - g_2 k \cos kl \]  (20)

\[ g_3 = \frac{jz_2}{k} - g_2 (f_1 \sin kl - f_2 \cos kl) \]  (21)

2.2. The Transfer four End Parameters of the Tapered Pipe with the Reverse Structure

When the structure is the four end parameters' calculation method of the longitudinal vibration's transfer characteristics is given in the above sections. In fact, the structure may be under reverse incentives, the input and output positions of the transfer matrix need to be swapped, the inverse matrix of the structural vibration's transfer four end parameter matrix need to be solved to obtain the reverse transfer four end parameter matrix of the structure.

Assuming the forward transfer four end parameter relation of the structure is:

\[ \begin{bmatrix} v_1 \\ F_1 \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix} \begin{bmatrix} v_2 \\ F_2 \end{bmatrix} \]  (22)

If the structure is under reverse incentives, the transfer four end parameter can be wrote:

\[ \begin{bmatrix} v_2 \\ F_2 \end{bmatrix} = \begin{bmatrix} b_{11} & b_{12} \\ b_{21} & b_{22} \end{bmatrix} \begin{bmatrix} v_1 \\ F_1 \end{bmatrix} \]  (23)

The two transfer matrixes satisfy the following relation:

\[ \begin{bmatrix} b_{11} & b_{12} \\ b_{21} & b_{22} \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix}^{-1} \]  (24)

3. THE DESIGN OF THE ACOUSTIC ISOLATOR

3.1. The Design of the Variable Cross Section Acoustic Isolator

According to the standard of drill pipe, the acoustic isolator is design as the follows structure, as shown in Fig. (6), the output end is on the left, while the input end is on the right. Both ends of acoustic isolator is the NC50 standard thread interface, the length of each side is different, the length of the left interfaces is 254mm, the radius is 89mm, the length of the right interfaces is 292.1mm, the radius is 89mm. The hollow straight cone pipe is used to connect the Interfaces with the central cylindrical pipe, the length of the left straight cone pipe is 80mm, and the length of the left straight cone pipe is 70mm. The length of the middle small section cylinder pipe is 3245.4mm, the section radius is 63.5mm, The length of the larger section cylinder pipe is 1500mm, the section radius is 89mm. The whole length of the acoustic isolator is 8686.8mm, the maximum section radius is 89mm.

When calculating the insulation body is divided into seven sections with different cross section hollow cylindrical rod and cone rod combination, using the theory of structure vibration, each of the section's structure design parameters are put into the program for the vibration transmission characteristics structure of the structure, the result is as show in Figs. (7, 8). Here needed to explain: considering the thread interface will be docking with the before and after the drill pipe, and the interface will be a uniform cylindrical pipe after the combination, therefore here the NC50 thread interface is calculated as a uniform cylindrical pipe.
5 orders of magnitude, amplified phenomenon is obvious. Fig. (8) is the velocity admittance curve of the structure's input force under the free boundary condition, and it is always the speed response of the output end with the input force of 1N. Which the maximum velocity occurs in the 302Hz, the value is 0.053m/s. The corresponding frequency points of the rest vibration peaks are: 451Hz, 923Hz, 1051Hz, 1505Hz, 1735Hz, 1974Hz, 2439Hz etc.

3.2. The Finite Element Simulation of the Variable Cross-section Acoustic Isolator

In order to verify the design, using the finite element software ANSYS, the design model is simulated, the model is shown in Fig. (10), it is respectively calculated natural frequency of the structure which is showed in Table 1, and natural mode of vibration which is showed in Fig. (11), and got the vibration structure transfer response with the unit force excitation.

![Fig. 7](image1.png)

**Fig. (7).** The transfer coefficient with the output end free and fixed boundary conditions.

![Fig. 8](image2.png)

**Fig. (8).** The transfer admittance curve with the condition of the output end free.

![Fig. 9](image3.png)

**Fig. (9).** The structural transfer admittance curve of the 500-1500 frequency band.

**Table 1.** The natural frequency of the acoustic isolator's first 8 orders.

<table>
<thead>
<tr>
<th>Order</th>
<th>Simulation value</th>
<th>Theoretical value</th>
<th>Error %</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>292.30</td>
<td>302</td>
<td>3.2</td>
</tr>
<tr>
<td>2</td>
<td>439.69</td>
<td>451</td>
<td>2.5</td>
</tr>
<tr>
<td>3</td>
<td>902.98</td>
<td>923</td>
<td>2.1</td>
</tr>
<tr>
<td>4</td>
<td>1025.9</td>
<td>1051</td>
<td>2.3</td>
</tr>
<tr>
<td>5</td>
<td>1485.8</td>
<td>1505</td>
<td>1.2</td>
</tr>
<tr>
<td>6</td>
<td>1701.4</td>
<td>1735</td>
<td>1.9</td>
</tr>
<tr>
<td>7</td>
<td>1950.5</td>
<td>1974</td>
<td>1.1</td>
</tr>
<tr>
<td>8</td>
<td>2402.3</td>
<td>2439</td>
<td>1.5</td>
</tr>
</tbody>
</table>

The natural frequency error of calculation results of and theoretical analysis of is not more than 3.2%, the minimum phase difference is 10Hz. From the inherent modal to see, the vibration displacement of the acoustic isolator's ends is the biggest at the structure resonant frequency.

The unit excitation force is applied to the acoustic isolator to calculate the transmission characteristics of structure vibration on free boundary condition, and acquire the node vibration velocity of the acoustic isolator's lower end, compare the differences between the calculation results of the structural transfer admittance's finite element simulation with the results of theoretical calculations, as shown Fig. (12).

The peak frequency of finite element calculation is consistent with the theoretical calculation results before 3000Hz.
Due to Response analysis of finite element considering the damping factor, theoretical calculation does not take into account the structural damping, so the finite element calculation values are slightly lower than the calculated in the peak size.

The pass band and stop band of the finite element calculation frequency are fully consistent with the theoretical calculation values, which also proves the reliability of this calculation method again.

CONCLUSION

In this paper the longitudinal vibration velocities and force expressions of several kinds of variable thickness cylindrical pipes and tapered pipes are given, according to the relationship between the input and the output ends, the four end parameters method of vibration transmission is established, the indexes of transfer admittance parameters and transmission coefficient defined in this paper are compared to reflect the vibration transfer characteristics of combined acoustic isolator's structure. Based on this method, a variable section cylinder pipe combination acoustic isolator is designed, the theoretical calculation of which is consistent with the finite element simulation results, the acoustic isolator finally can meet the sound insulation effect for 50dB in the band of 700-800Hz and 1200-1300Hz.
CONFLICT OF INTEREST

The authors confirm that this article content has no conflict of interest.

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Declared none.

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