Principles of developing insulators for acoustic logging tools

Gilmanova A M

Department of Mechanics and Machine Building Technology, Ufa State Petroleum Technological University, Branch of the University in the City of Oktyabrsky, Oktyabrsky, Republic of Bashkortostan, Russia

E-mail: alfiya_okt@mail.ru, info@of.ugntu.ru

Abstract. The paper describes the requirements for acoustic insulators, experimental and theoretical foundation of insulator structure, and proves the appropriate calculations. The work output is the modernized design of insulators that suppress interference waves between the transmitter and receiver of the downhole probe depending on the equipment used and the well under study.

1. Introduction

The paper “Consideration of various factors influencing acoustic logging equipment production accuracy” [1] thoroughly considers the factors affecting the measurement error of acoustic logging methods implemented by means of downhole instruments and analyzes the systematic components of these errors being the quality of equipment design and manufacture as well as its correct exploitation.

The present paper considers the following principle indicated in [1]: ensuring the interference waves suppression between the transmitter and receiver of the downhole probe.

When conducting acoustic logging on head waves, the waves carrying useful information are accompanied by the waves propagating from the transmitter to the receiver directly along the body of the tool regardless of its position in the borehole. These waves are noises because their propagation speed and energy can commensurate with useful waves. To suppress these interference waves, an acoustic insulator (hereinafter “insulator”) is placed between the emitter and the receiver. The insulator is calculated for a given frequency range and wave intensity and for each type of acoustic equipment depending on its area of application.

Difficulties in designing an insulator relate to the fact that an insulator must simultaneously satisfy the following conflicting requirements [2]:

- on the one hand, the insulator should provide maximum absorption of acoustic energy, i.e. the material it is made from must have high damping or low sound speed;
- on the other hand, it must have a sufficiently high mechanical tensile and compressive strength to prevent changes in the measuring base and the transducer spacing (which lead to unacceptable errors in measuring the temporal and amplitude parameters of the received signals), and sufficient lateral rigidity in the orientation over a borehole to exclude deviations in the position of the acoustic probe from a central location in inclined and horizontal wells.

Insulator designs available in our country and abroad can be divided into two large groups: tight and non-tight acoustic insulators.

Non-tight insulators include structures in which wires are routed from node to node through borehole fluid, which is often under high hydrostatic pressure and temperature [3, 4]. The disadvantage of such insulators is that along with a high efficiency of acoustic insulation such structures are complicated by
additional bridges for hermetically tight wire entry into the units of acoustic transducers. In addition, over time such structure gets clogged with sludge, which after drying turns into a solid filler degrading the insulating properties of the acoustic insulator.

In hermetically tight insulators, the wires between the acoustic transducer assemblies pass inside the borehole housing, which usually has the form of a cylinder with the grooves cut along a helical line; the grooves have different pitches and depths and can be located both on the inner and outer sides of the housing. The low speed of elastic waves propagation, increased attenuation and the required rigidity of such structures, whose outer grooves are filled with material having a high specific gravity and low modulus of elasticity, for example, lead, enable to successfully use them in acoustic devices of large diameters. However, the difficulty in being manufactured for high pressures (above 600 kg/cm) and reduced diameters (less than 48 mm) make such structures not applicable to small-sized devices [5, 6].

At the stage of design, it is necessary to remember that the operating conditions of insulators base on the operating conditions of small-sized downhole tools and are cased and uncased wells, i.e. the environments with extremely high (> 5,000 m/s) and extremely low P-wave speed (<2,000 m/s), as well as an aggressive environment being methane, hydrocarbons, hydrogen sulfide, oil, and other ones.

2. Methods and materials
The results obtained are briefly presented in the paper and base on the analysis of theoretical data, calculations, experimental data and well studies.

3. Content of research and evaluation of its efficiency
Due to the complexity and importance of developing an acoustic isolator for the parametric series of small-sized acoustic logging equipment (and we consider the development of insulators for such a series [1]), it is necessary to clarify the main points of the theory of elastic waves propagation in geometrically bounded media.

The attenuation of elastic waves is predetermined, on the one hand, by the type of waves, and on the other, by the physical and mechanical characteristics of the medium. The attenuation is considered by introducing the imaginary component into the expression for the wavenumber \( k = \frac{2\pi}{\lambda} + j\delta \) (\( \delta \) is the attenuation coefficient). It is known from [7] that due to diffraction divergence, the amplitude of bulk longitudinal and transverse waves decreases according to the \( 1/kr \) law, while the amplitudes of the Rayleigh, normal and diffracted waves decrease according to the \( 1/\sqrt{kr} \) law, and the head amplitude decreases according to \( 1/(kr)^3 \). In the general case, the attenuation coefficient is the sum of the absorption coefficients \( \delta_a \) and scattering coefficients \( \delta_s \), i.e. \( \delta = \delta_a + \delta_s \).

Upon absorption, the acoustic energy flux passes into a heat flux, and upon scattering it remains acoustic but escapes through the directionally propagating beam. Sound absorption is due to internal friction and thermal conductivity of the medium. For the same medium, the absorption of transverse waves is less than that of the longitudinal ones since they are not associated with adiabatic changes in volume, at which losses for thermal conductivity appear. The absorption coefficient in solids is proportional to either \( f \) (glass, metals) or \( f^2 \) (rubber). Absorption is the dominant factor causing ultrasound attenuation in single crystals.

Metals applied practically have a polycrystalline structure; and the waves attenuation in them is predetermined by two main factors being reflection and scattering of ultrasound due to the anisotropy of the mechanical properties of the metal. As a result of reflection, the front of the ultrasonic wave deviates from the rectilinear direction of propagation, and the amplitude of the received signals drops sharply. In addition to reflection, a wave incident on the boundary of crystals (grains) undergoes partial reflection, refraction of ultrasound and transformation, which determines the scattering mechanism. For coarse-grained materials (copper, corrosion-resistant steel), both phenomena are characteristic, as a rule, while in fine-grained materials (low-carbon steel, aluminum) scattering predominates.

When properly scaled, this theory applies to the operation of an acoustic isolator. The fraction of the dissipated energy is mainly determined by the ratio of the elastic wavelength to the average crystallite size, i.e. \( D \) denotes details in our case. At \( \lambda = D \), ultrasound scattering is very large, and it is maximum in the range \( \lambda/D = 3...4 \). This is the area of diffuse scattering. At \( 4 \leq \lambda/D \leq 10 \), the scattering coefficient
is proportional to the product D², and at λ/ D > 10, it is proportional to D¹. The smallest attenuation is observed at λ/ D ≥ (20...100).

Thus, the structure of an acoustic insulator based on the common principles of alternating media with sharply differing acoustic impedances and lengthening the trajectory of sound is nothing more than the principle of wave attenuation due to refraction and scattering of ultrasound. In this case, it is necessary to consider the elements of the acoustic insulator, whose dimensions should correspond to the ratio λ/ D = 3...4 instead of the crystals with their sizes.

With an oblique incidence of a longitudinal wave at an angle α from one solid medium to the boundary with another solid medium, the wave is reflected, refracted and transformed, and in the general case, four more waves arise (Figure 1) and they are as follows: two refracted P and S (velocities C₁ and C₅) and two reflected P and S (velocities C₁ and C₅) ones [2].

![Figure 1. Scheme of reflections, refractions and transformation of waves at oblique incidence from one medium to another](image)

All waves directions lie in one plane being the plane of incidence formed by the incident ray and the normal to the reflecting surface reconstructed at the point of the ray incidence.

The waves propagation times in the first and second medium are equal to each other. Snell’s law (or the law of sines) operates between the angles and the corresponding speeds:

\[
\sin \alpha / C_1 = \sin \alpha / C_{l2}
\]

(1)

The angle at which the refracted P-wave merges with the boundary (β=90⁰) and becomes inhomogeneous is called the first critical angle α₁. The angle at which the refracted S wave disappears (β= 90⁰) is called the second critical angle α₂. When a transverse wave is incident on the interface between the media, it is possible for the longitudinal reflected wave to be absent; the angle for this case is called the third critical angle α₃=arcsin C₅/C₄. For steel α₃ = 33.5⁰. The ability of ultrasound to reflect from the interfaces between media with different acoustic impedances is characterized by the reflection coefficient R (in amplitude), which is the ratio of the pressure amplitudes in the reflected and incident waves:

\[
R = P_{ref}/P_0 = \left(\frac{\rho c}{\cos \beta} + \frac{\rho_{l1} c_1}{\cos \alpha} \right) / \left(\frac{\rho c}{\cos \beta} + \frac{\rho_{l1} c_1}{\cos \alpha} \right)
\]

(2)

In this case, the amplitude transmission coefficient D = A_{trans}/A₀.

Formulas for calculating the reflection and transmission coefficients in energy in the case of two solids or a liquid and a solid were obtained by D.V. Dianov by a rigorous solution of the problem at the boundary between two media under the conditions of equality of normal and absence of tangential stresses. With a direct fall,

\[
R = \left(\frac{Z \cdot Z_1}{Z + Z_1}\right)^2,
\]

(3)

\[
D = \left[4ZZ_1 / (Z + Z_1)^2\right]^2,
\]

(4)

where Z is impedance being the ratio of the acoustic pressure to the normal component of the vibrational velocity, due to which energy is transferred from one medium to another:

\[
Z = P_0 / U_v,
\]

(5)
Regarding the ratios of the quantities characterizing the plane harmonic wave:

\[ \nu = jk\phi; \quad \rho = j\omega\rho, \]  

(6)

where \( j = k = n\omega/c \) is a wave number \( k = \omega/c = 2\pi/\lambda \), \( n \) is a unit vector characterizing the wave direction; \( \omega = 2\pi f \) is angular frequency; \( f \) is vibration frequency; \( f = Fe^{j(kr - \omega t)} / r; r \) is the distance from the origin; \( k_x = k_y = k_z \)

we get:

\[ P_n / \nu = \rho c. \]  

(7)

The normal component of the vibrational velocity of the incident wave is as follows:

\[ \nu n = \nu \cos \theta_{inc}. \]  

(8)

Substituting (7) and (8) into (9), we have

\[ Z = \rho c / \cos \theta_{inc}, \]  

(9)

where \( \theta_{inc} \) is angle of wave incidence \( \alpha \).

With an oblique incidence of a longitudinal wave:

\[ R = \left( \frac{Z_c \cos^2 2\alpha + Z_s \sin^2 2\alpha - Z_t}{Z_c \cos^2 2\alpha + Z_s \sin^2 2\alpha - Z_t} \right)^2 \]  

(10)

\[ D_1 = \frac{4Z_cZ_s \cos^2 2\alpha}{(Z_c \cos^2 2\alpha + Z_s \sin^2 2\alpha + Z_t)} \]  

(11)

\[ D_2 = \frac{4Z_cZ_s \cos^2 2\alpha}{(Z_c \cos^2 2\alpha + Z_s \sin^2 2\alpha + Z_t)} \]  

(12)

The results of calculating the transmission coefficients of the longitudinal \( D_1 \) and transverse \( D_2 \) waves through energy depending on the angle of incidence according to formulas (11) and (12) are shown in Figure 2, the optimal dimensions of the acoustic insulator parts depending on the frequency of the emitters are shown in Figure 3, the dependence of the reflection coefficient on the angle of incidence is shown in Figure 4.

In the area with incidence angles above 28° (Figure 2), there is practically only a surface wave, which is logical to direct into the inner cavity of the insulator, into the air in the structure of insulators for acoustic instruments for full wave logging. It is stipulated by the fact that the waves attenuation in the air is large, and the output of interference waves into the borehole fluid gives additional reflected-refracted waves, which negatively affect the quality of information with the full wave AK method.

It should be noted that in a certain range of angles the theoretical values of the transmission coefficients differ from the experimental ones. This leads to the discrepancy between the actual results and Snell’s law. For example, \( \alpha = 30^\circ, \beta_i = 37^\circ, \beta_{exp} = 39^\circ, \alpha = 53^\circ, \beta_i = 76^\circ, \beta_{exp} = 72^\circ \), where \( \beta \) is the angle of incident calculated according to Snell’s law, \( \beta_{exp} \) is the experimental angle of incident. The difference is explained by the fact that Snell’s law is valid for plane waves, while the beam in real transducers, as a rule, diverges, and each of the beam’s rays has its own divergence coefficient.

It turns out that in order to improve the insulating properties of the acoustic insulator, a layer of rubber or fluoroplastic must have such a thickness at which the maximum reflection and the minimum transmission of waves can be achieved:

\[ h_{lay} = 0.25 (2n-1) \lambda_c. \]  

(13)
where $h_{lay}$ is layer thickness; $\lambda_{lay}$ is elastic wavelength in the layer material.

For a very thin layer $h_{lay} \ll \lambda_{lay}$, on both sides of which there are identical media with impedances $Z=Z_1$, the following formula is valid:

$$R = \frac{1}{1 + Z_{lay} \lambda_{clay} / (\pi h_{lay} Z)^2}$$

(14)

Fig. 2. Dependence of transmission coefficient of elastic waves on their incidence angle

Fig. 3. Dependence of optimal length of acoustic insulator parts on material used and operating frequency

F is fluorine plastic, R is rubber, S is steel
Fig. 4. Dependence of reflection coefficient on angle of incidence

Relation $\rho' c' / \rho c = \zeta$ is relative wave impedance.

If $\zeta = 1$, the reflection coefficient is zero and the wave normally incident on the interface between the two media passes from medium I to II entirely without being reflected.

If $\zeta > 1$, the reflection coefficient is positive.

If $\zeta \to \infty$, the reflection coefficient tends to 1.

If $\zeta$ is very small (i.e. the second medium is acoustically very soft compared to the first one), the pressure tends to zero, and the particle velocity tends to double the velocity in the incident wave (I).

If $\zeta$ is very large, the velocity of particles at the boundary tends to zero and the pressure doubles.

The reflection coefficient in absolute value is always less than one: it is positive if the wave falls from a medium with a lower characteristic impedance and negative in the opposite case.

Analysis of the previous theoretical material and the corresponding mathematical calculations, whose results are summarized in tables 1 and 2, enables to conclude that the acoustic insulation in the acoustic logging equipment can be considered satisfactory if at least one of the following conditions is met [2]:

1) the amplitude of the forward wave along the casing or the amplitude of the wave associated with the body is several (5-7) times weakened in comparison with the minimum signal; 2) the delay time of the direct wave along the casing is so long that it does not fall into the longitudinal wave recording material at all values of the recorded velocities.

Table 1. Calculated critical angles

| Section boundary of insulator parts | $\alpha_{kp1}$ | $\alpha_{kp2}$ | $\alpha_{kp3}$ | $m = \frac{\rho_1 C_2}{\rho_2 C_1}$ |
|------------------------------------|----------------|----------------|----------------|----------------------------------|
| rubber-steel                       | 14°39'         | 27°16'         | 0              | 0.029                            |
| fluorine plastic-steel             | 14°27'         | 26°50'         | 25°10'         | 0.07                             |
| air-steel                          | 3°14'          | 5°52'          | 0              | 0.03                             |
| water-steel                        | 14°45'         | 27°29'         | -              | 0.033                            |
Table 2. Selection of optimum length of structural elements

| Insulator part material | Radiation frequency, kHz | Wavelength, m | Length of structural parts of acoustic insulator, mm |
|-------------------------|--------------------------|---------------|------------------------------------------------------|
|                         |                          | n=1 | n=2 | n=3 |
| steel 30 HGSA           | 6                        | 0.97 | 243 | 729 | 1215 |
|                         | 12                       | 0.49 | 122 | 366 | 610  |
|                         | 18                       | 0.33 | 80  | 240 | 400  |
|                         | 20                       | 0.29 | 73  | 219 | 366  |
|                         | 6                        | 0.24 | 62  | 186 | 310  |
| rubber                  | 12                       | 0.12 | 31  | 93  | 155  |
|                         | 18                       | 0.08 | 21  | 62  | 105  |
|                         | 20                       | 0.07 | 18.5| 56.5| 92.5 |
|                         | 6                        | 0.24 | 61  | 183 | 305  |
| fluorine plastic        | 12                       | 0.12 | 31  | 91  | 152  |
|                         | 18                       | 0.08 | 21  | 61  | 105  |
|                         | 20                       | 0.07 | 18  | 55  | 92   |

Analysis of the available domestic and foreign literature and the existing variety of designs of acoustic insulators enables to conclude that when developing an acoustic insulator for equipment intended for use in horizontal wells and horizontal sidetracks, it is necessary to consider the following information [9]:

1. There is a dependence of the elastic wavelength on the material of the insulator parts and the operating frequency of the converters (see Table 1);
2. The signal propagating along the axis of the insulator undergoes a strong attenuation due to multiple reflections at the boundaries of media with sharply different acoustic stiffnesses, and the greater the difference in stiffness, the greater the attenuation is [10];
3. When waves are reflected and refracted on a conical interface with different taper angles, energy is redistributed between the waves of different types (P and S), whose distribution velocities are significantly different, as well as lengthening of the total path of elastic waves inside the insulator caused by refraction and reflection of waves at different corners;
4. The arrangement of interfaces between elements with different acoustic stiffnesses in the structure should satisfy the following requirement: waves refraction and reflection on these surfaces must be directed inside the volume of the insulator, i.e. the cones tops must face the receivers;
5. The weakening of acoustic bonds (transient acoustic energy losses at the contacts of different layers) is inversely proportional to the area of the contacting surfaces;
6. The effect of “high-frequency filtration” consists in the fact that for some substances such as viscous bodies (plasticine, water, soap, epoxy resins liquid, Mendeleev putty, rubber, bitumen) forming several layers or rods, with a decrease in the thickness of each layer to some value (6-4 mm), there is an increase in the visible frequency of the propagating elastic wave and, consequently, an increase in its attenuation, a decrease in the amplitude of the elastic pulse [11]. It is also known that if the thickness of the housing wall is much less than the wavelength, then damping increases (due to the predominant excitation of flexural waves).
Selective absorption of low frequencies is due to the transformation of bulk waves into Lamb waves, which exhibit strong absorption of low-frequency components of the spectrum under certain conditions.

7. The structure of the isolator should be sufficiently rigid in orientation along the borehole axis and exclude deviation of the position of the acoustic probe from the central location in the borehole, i.e. optimal transverse rigidity is required to maintain the shape of the device in the controlled directional wells, horizontal wells [12, 13].

8. The structure of the insulator must have sufficient axial rigidity to keep the measuring base and the spacing of the transducers unchanged in the probe, which leads to unacceptable errors in measuring the time and amplitude parameters of the received signals.

9. The presence of open metal surfaces in the structure of the insulator causes the appearance of reflected-refracted waves distorting the wave pattern, which is unacceptable when registering a full wave acoustic signal [14].

Based on theoretical calculations and experimental findings, as well as the available domestic and foreign experience in the construction of acoustic insulators, the author has developed several options for tight and non-tight insulators (Figure 5).

![Figure 5](image)

**Fig. 5.** a is option 1, b is option 2, c is option 3, d is option 4, e is option 5

It was decided to test five options of acoustic isolator structure (three options of hermetically tight insulators and two – non-tight ones) in the complex small-sized acoustic logging equipment included in the parametric range.

The first option of a tight pipe-shaped insulator with through grooves and double (external and internal) rubber coating (Figure 5, a), was used in the structure of the equipment with a diameter of 48 mm. The selected rubber coating, on the one hand, ensures a better tightness of the structure, and, on the other, improves its insulating properties. Additional resistance against external hydrostatic pressure is
provided by filling the inner cavity with a liquid dielectric (transformer oil, organosilicon liquid) and using an elastic element functioning as a pressure compensator.

Another hermetically tight insulator (Figure 5, b) was a type-setting structure that allowed changing its length, if necessary.

The third option of the tight insulator (Figure 5, c) consisted of series-connected identical short nodes rubberized into a single block.

The fourth type of insulator being a non-tight one (Figure 5, d) with a diameter of 36 mm was made of fluoroplastic parts with internal transverse grooves, into which steel washers were inserted to increase the lateral rigidity, and metal cups acting as bandages were put on the outside.

The fifth option (Figure 5, e) was presented in the form of a rubberized assembly of figured plates (with inserted half washers) welded to the half bushings.

The method for calculating insulators, which must be addressed in order to develop them is given in various sources [15, 16].

The effect of mutually overlapping transverse grooves of the first option of the acoustic insulator depended on the coefficient of grooves overlapping and the distance of one groove from another. However, it is logical that the insulator casing itself must have sufficient mechanical strength in tension and bending. To provide this, it was necessary to perform the following calculations for mechanical strength: tensile strength of at least 3/4 of the breaking force of the cable; bending not less than 10⁴ kg/cm², as well as an acoustic calculation, which consisted in determining the minimum path of the elastic wave when bending around the grooves; the main task was considered to obtain the maximum elongation of the path with sufficient mechanical strength [18, 19].

In general, the mechanical calculation of the rest of the acoustic insulators is as well reduced to the choice of the optimal option between the required mechanical strength of the assembly and the maximum possible lengthening of the path of elastic waves through the assembly elements with regards to the theoretical material described.

The tests of the developed samples of insulators to compare their characteristics were carried out by connecting them to the same device with unchanged acoustic emitters and a receiver, and by measurements conducted in a test tube filled with water, and then in a bench well of OAO NPP “VNIIGIS” [1].

Tests of the developed structures of acoustic insulators showed the following:
1. With a high efficiency of acoustic insulation, the fluoroplastic parts of the non-tight insulator (option 4) turned out to be insufficiently heat-resistant: after several wells with a maximum temperature T>100°C, the cylindrical parts lost their shape. In addition, at a well depth of more than 1,000 m, the effectiveness of their isolation decreased sharply apparently due to the phenomenon of compaction of fluoroplastic material caused by all-round hydrostatic compression;
2. High efficiency with unchanged layout and the same length of insulators was obtained by a non-hermetic rubber insulator (option 5); the disadvantage of such structures is the need to dismantle the insulator after each well to clean and flush the inner cavity of the insulator from cuttings;
3. A tight insulator in the form of a pipe with grooves and double rubber (option 1) also had a high insulating efficiency. Moreover, an important advantage of this structure was the good reproducibility of the acoustic characteristics of the insulator from sample to sample. However, the long-term practice of using such a design has revealed its weakness associated with the fragility of the rubber material, which filled the insulator grooves. The service life of the rubberized insulator is from 7 to 8 years (although the service life of the downhole tools themselves according to the technical conditions is 6 years, in production they work 2-3 times longer). The tightness of the rubberized insulator is provided only by rubber lining, and when the rubber “grows old”, it loses elasticity, and, accordingly, resistance to temperature and pressure, then the characteristics of such an insulator deteriorate sharply;
4. Type-setting insulator (option 2), whose tightness does not depend on the external rubber (or fluoroplastic) parts, is most promising in wells with an aggressive environment. The necessary insulating effect of an acoustic insulator of such structure while ensuring its small size is achieved due to the following factors:
   - determining the length sizes of alternating elements, as well as the total length of the acoustic insulator, which vary depending on specific conditions, on the basis of calculations and confirmation by experiments of the optimal ratios;
- using the same method to select angles formed by transverse conical surfaces with a plane perpendicular to the axis of the insulator in cross section lying outside the second critical angle; moreover, the tops of these corners are directed towards the receiver.

According to the results of bench and borehole tests, a hermetically tight rubberized acoustic insulator (option 3) consisting of repeated short bushings and enabling to change the length of the insulator and the layout of the downhole tool probe if necessary proved to be the most effective (Table 3).

Table 3. Test results of acoustic insulators

| Insulator option | Figure | Length, mm | Diameter, mm | Speed, v, m/s | Rel. ampl., Aiz / Aet |
|------------------|--------|------------|--------------|---------------|----------------------|
| 1                | 2.9 a  | 285        | 48           | 570           | 0.3                  |
| 2                | 2.9 b  | 296        | 36           | 705           | 0.5                  |
| 3                | 2.9 c  | 240        | 36           | 500           | 0.23                 |
| 4                | 2.9 d  | 296        | 36           | 423           | 0.43                 |
| 5                | 2.9 e  | 296        | 36           | 657           | 0.16                 |
| etal. pipe       | –      | 296        | 36           | 3600          | 1                    |

4. Conclusion

Theoretical and experimental studies carried out considering the formulated principles and requirements were used as the basis for finding original technical solutions for the structures of acoustic insulators providing attenuation up to 80 dB/m and a delay in the arrival time of a wave along the casing equal up to 400 \( \mu \text{s/m} \).

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