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Analysis of the Impact of Energy Losses in Operating Components on the Performance of Hydraulic Mining Excavators

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Abstract: The use of hydraulic excavators in open pits has a high force effect on the teeth of the bucket, reduces the operating cycle compared to mechanical excavators by 15-18%, which in turn increases operational efficiency by 30-35%.
The hydraulic system is the mechanism that performs the excavation period of the hydraulic excavator, the period of bucket rise, the period of torsion of the loaded bucket, the period of loading, the period of torsion of the unloaded bucket, the period of unloading the bucket. Its optimal performance depends on the design of the hydraulic equipment, the technical condition of the hydraulic excavator and the working environment.

Keywords: Hydraulic excavator, loading period, excavation period, pressure energy loss.

1. Introduction

The most commonly used type of digging-loading machines, which are used in the work of digging in the open method, are single-sink career excavators.

A machine designed to sink the mining mass (excavating), moving it for a relatively long distance and loading it into vehicles or overhangs, is called an excavator.

The working cycle of a single sink excavator consists of four consecutive operations: filling the sink (immersion), moving it to the place of discharge (transportation), unloading and silencing the empty sink to the place of immersion in order to carry out the new cycle. Therefore, a cycle of excavators with a sink (continuous) is considered a moving machine.

As a result of the scientific research and practical application of manufacturing enterprises and several scientists, the following main indicators affecting the general working state of gravity excavators are shown [1]:

- Pinch of salt;
- Base-handle length;
- Handle length;
- Theoretical productivity;
- Working time before the first capital repair;
- Total weight;
- Energy consumption.

From the above indicators, we will analyze the cases of dependence on the power system during operation on the basis of the impact of malfunctions.

As we know, the indicators of energy loss in the power transmission circuits of gravitational excavators will depend on its productivity. There are 3 different types of productivity of hydraulic excavators and they are theoretical, technical and operational [2].

Theoretical productivity, m^3/sec

$$Q_n = 60qn = q\frac{3600}{T} \qquad (1)$$

Where, q-the size of the cowl, m^3; n - the number of working cycles per 1 hour; t - the duration of the cycle, sec;

Theoretical productivity, m^3/ s

for the reverse bucket $T_{rev.buc} = t_q + t_k + t_b^I + t_y + t_b^{II} + t_t$

for the direct bucket $T_{dir.buck} = t_{zab.kir} + t_{to\prime l} + t_k + t_b^I + t_y + t_b^{II}$ (2)

where, t_q – excavation period, sec; t_k – time for the bucket to rise, sec; t_b^I– loaded bucket turning time, sec; t_y – load release time, sec; t_b^{II} – load-free bucket turning time, sec; t_t – the time of the descent of the bucket into the slaughter, sec.

For excavators with a reverse bucket.

$t_{zab.kir}$– time to get rid of effortlessly hydraulic cylinder into slaughter, sec; $t_{to\prime l}$– the filling time of the hopper in the bend at an angle of 45^0 degrees, sec; t_k– the time of bucket ascent, sec; t_b^I– the turning time of the loaded bucket, sek; t_y– load release time, sec; t_b^{II} – the turning time of the load-free bucket, sec.

Technical productivity for flat bucket excavators, m^3/s

$$Q_t^I = 60qnk_t \frac{1}{k_y} k_q \quad (3)$$

Where, k_t– filling coefficient of the bucket; k_y– coefficient of loosening of the rock; k_q– coefficient under the influence of difficulty in digging the rock;

Operational productivity, m^3/sec

$$Q_e = Q_t^I k_v = 60qnk_t \frac{1}{k_y} k_q k_v \quad (4)$$

Where, k_v- the coefficient of taking into account the level of use of the excavator over time.

This means that we can see that all of the above indicators are affected by the above productivity. We have taken note of the fact that one of these indicators is of particular importance the excitation cycle. The indicators remaining until the bypass this process is also very important. But we considered that the process of influencing the work activity of an entire excavator of the hydraulic system as the most optimal indicator in the study of the impact of both the survey and the generated malfunctions. Because in the excavating and loading processes in the excavating cycle, we can see the processes that take place in the system as a result of the execution of a large number of power hydro motor action phases.

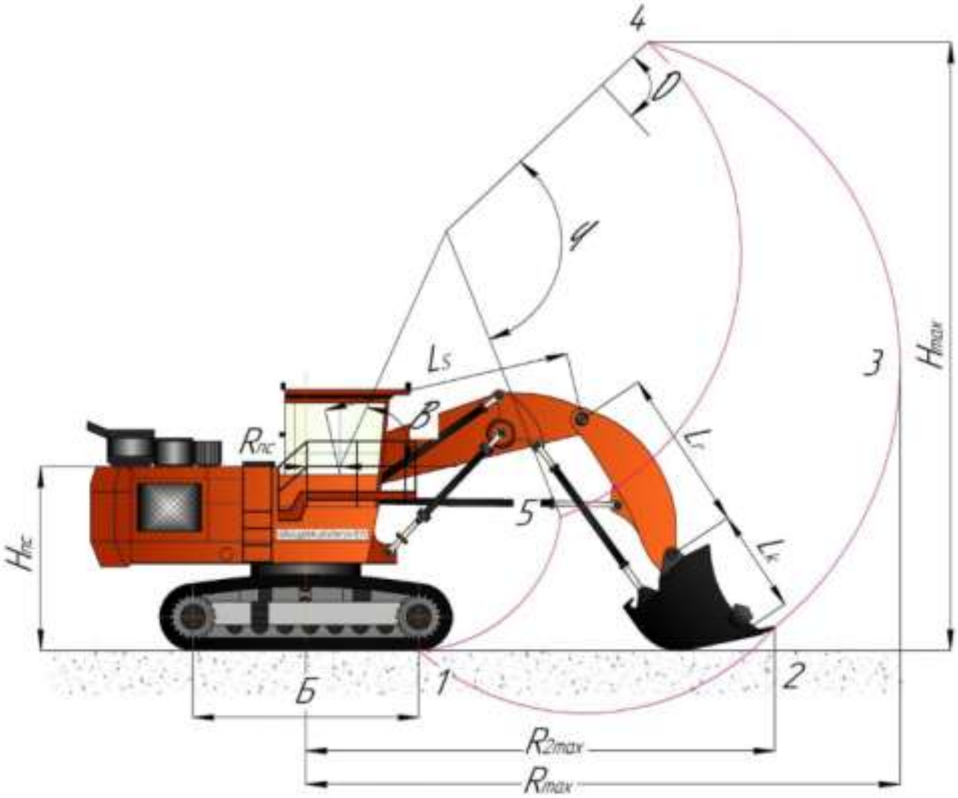
In the research of excavation-loading work on excavators, we can see the epigraphic curvature formation of a straight–wheeled excavator at 5 points (Figure 1). The developed slaughter impresses the excavation plot trajectory, that is, it involves the formation of geometrical laws. For the formation of the excavation curvature line plot, the excavator begins at 1-point lower than the standing plane, at this point the excavator moves to the 2-point sink, starting at B/2 from half equal to the rotation axis of the base located at the Standing level. The maximum ledge of cleaning this splash bottom is R_{2max}. 1-point to 2-point digging plot trajectory is considered a logarithmic spiral, at 1-point the support cylinder and the sink are in a closed position, that is, the angle of rotation of the handle γ=0, the angle of inclination of the sink θ=0, the angle of ascent of the base-handle β=0 will be equal to.

At the 2-point base-handle, the handle begins to cylinder the bucket to the 3-point, where the rollers of the sinks are in the moving position. The point is H_{PS} at the height below the excavator, and the maximum distance from the axis of rotation of the excavator is the maximum digging radius R_{max}.

In this case, the shaft of the handle cylinder will be the maximum output, the angle of rotation of the handle will be equal to γ=max, and the tray will come to the logarithmic spiral position.

Moving the bucket to point 4 reaches the maximum digging height of H_{max}. The shaft of the boom-lever cylinder is in the maximum protruding position, and the angle of lifting of the boom-lever is β = max. The radius of the notch is found by subtracting the distance difference from the maximum radius of the notch to the axis of rotation of the boom (R_{max}-R_{PS}).

The digging trajectory, which is 1-4, will be equal to the maximum operating parameter of the hydraulic excavator.



1 - Picture. Working profile of a straight-buckled excavator in the field: β– boom lifting angle; γ– angle of rotation of the handle; θ– bucket lifting angle; L_s– boom height; B– basic height of hydraulic excavator; R_{max}– maximum digging radius; R_{2max}– maximum radius cleaning the sole of the ledge; H_{max}– maximum digging radius;

It has a minimum radius of recess from the axis of rotation of the hydro excavator and forms a trajectory at points 1,5,4. When moving from point 1 to point 5, the angle of rotation of the handle will be equal to $\gamma = 0$, while the angle of lifting of the handle-base varies from $\beta = 0$ to $\beta = max$. When moving the bucket from point 5 to point 4, the angle of rotation of the handle is carried out in the position $\gamma = 0$ to $\gamma = max$, and the angle of lifting of the boom-lever is in the position $\beta = max$. As a result, a plot of the excavation of a hydraulic excavator with a straight bucket is constructed.

The useful power output of the working devices of quarry hydraulic excavators will depend on the correct choice of the internal combustion engine unit and hydraulic system (Fig.2) [1].

Table - 1. Stages of sequential energy transfer in parts and mechanisms

Designation	Name of energy transition stages
X1	Output of the amount of energy from the internal combustion engine
X2	Gearbox
X3	The output energy of the gearbox
X4	Hydraulic pumps and auxiliary devices
X5	Output energy from hydraulic pumps and auxiliary devices
X6	Hydraulic pipes
X7	The amount of energy flowing through the hydraulic pipes
X8	Hydraulic valve
X9	Power output of the hydraulic spool valve
X10	Hydro cylinders
X11	Energy flow from the hydraulic cylinder to the implement

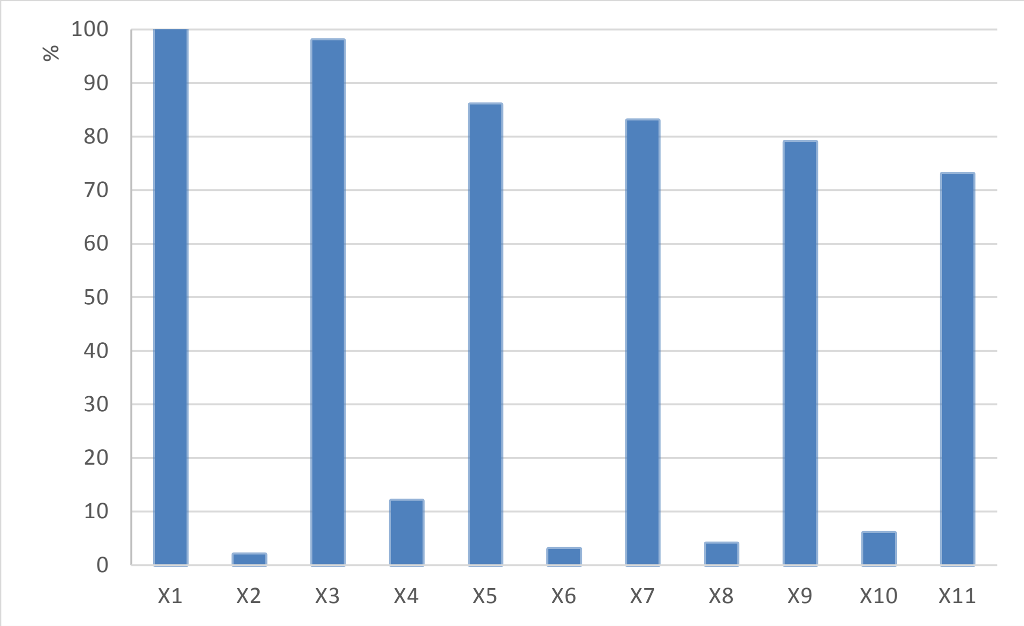


Figure 2: Diagram of the change in energy flow through the assemblies and mechanisms.

2. Materials and Methods:

In digging mode at point 1 to point 3 the highest load is created, at which point the lever hydraulic systems begin to move. As the lever cylinder moves, the hydraulic system working fluids experience some resistance as a result of the movement through the pipes. Determine the total pressure loss in the system up to the arm in the first case using the formula:

$$\sum \Delta P_{um} = \sum \Delta P_{ishq} + \sum \Delta P_{ul.q} + \sum \Delta P_{gid.el} \quad (5)$$

Where, ΣP_{um}– total pressure drop; $\sum \Delta P_{ishq}$ – Pressure losses for the length of rigid pipes and high-pressure hoses in the hydraulic system (service liquid and pressure absorption lines). $\sum \Delta P_{ul.q}$– Energy losses at connecting parts (connectors, sudden expansion, sudden contraction, fittings, couplings, flanges, etc.) in the hydraulic system. The local losses of each of these parts are calculated as 1.5 on average. $\sum \Delta P_{gid.el}$ – Pressure losses in the system due to series and parallel connected elements [3].
Determine the pressure drop resulting from the movement of the hydraulic cylinder rod in the crank arm using the formula [4]:

$$\sum \Delta P_{ruk} = \sum \Delta P_{ishq} + \sum \Delta P_{ul.q} + \sum \Delta P_{gid.el} \quad MPa \quad (6)$$

Where, $\sum \Delta P_{ishq}$ – pressure losses along the length of the pipe (discharge, suction and filler pipes). The operating-liquid velocity ϑ depending on the pressure and use of the hydraulic piping system is selected according to the practical suggestions developed [3].

- for suction pipe-1,0 ÷ 2,0 m/s;
- for filler pipe-1,5 ÷ 2,0 m/s;
- for the discharge pipe-4 ÷ 10 m/s.

$$\sum \Delta P_{ishq} = \Delta P_{sur} + \Delta P_{bos} + \Delta P_{dr} = P_{B\to N} + P_{N\to F} + P_{F\to R} + P_{R\to Gt} + P_{Gt\to P} + P_{P\to M} + P_{M\to Gs} + P_{Gs\to M} + P_{M\to P} + P_{P\to Gt} + P_{Gt\to B} = \frac{\lambda_{110}*L_{B\to N}}{d_{110}} * \rho * \frac{\vartheta_{sur}^2}{2} + \rho * \frac{\vartheta_{bos}^2}{2} * (\frac{\lambda_{31}*L_{N\to F}}{d_{31}} + \frac{\lambda_{38}*L_{F\to R}}{d_{38}} + \frac{\lambda_{31}*L_{R\to Gt}}{d_{31}} + \frac{\lambda_{31}*L_{Gt\to P}}{d_{31}} + \frac{\lambda_{38}*L_{P\to M}}{d_{38}} + \frac{\lambda_{31}*L_{M\to Gs}}{d_{31}}) + \rho * \frac{\vartheta_{dr}^2}{2} * (\frac{\lambda_{31}*L_{Gs\to M}}{d_{31}} + \frac{\lambda_{38}*L_{M\to P}}{d_{38}} + \frac{\lambda_{31}*L_{P\to Gt}}{d_{31}} + \frac{\lambda_{51}*L_{Gt\to B}}{d_{51}}) MPa \quad (7)$$

Where, ΔP_{sur}– total pressure loss over the length of the suction pipe; ΔP_{bos}– total pressure drop along the length of the discharge pipe; ΔP_{dr}– total pressure loss per length of pipeline to be poured; $P_{B\to N}$– pressure drop over the distance from the tank to the pump; $P_{N\to F}$– pressure loss from the pump to the high-pressure filter; $P_{F\to R}$– pressure loss from high-pressure filter to rotor control valve 6/3; $P_{R\to Gt}$– pressure loss from rotor control valve 6/3 to rotor control valve 8/3; $P_{Gt\to P}$– pressure drop from 8/3 hydraulic spool valve to over the boom plate; $P_{P\to M}$– pressure drop across the pipe cross-section from plate to coupling; $P_{M\to Gs}$– pressure drop from the clutch to the arm cylinder; $P_{Gs\to M}$– pressure drop from the hydraulic cylinder arm to the coupling; $P_{M\to P}$– pressure loss along the length of the pipe from the coupling to the plate handle; $P_{P\to Gt}$– pressure drop from plate to spool valve 8/3; $P_{Gt\to B}$– pressure loss from spool valve 8/3 to tank; n- and numbers in indexes STATE STANDARD (UZB) 6286-73, GOST 25452-17 high pressure pipeline hoses type n=31,38,51,110. λ_n- is the coefficient of hydraulic friction. To find the hydraulic friction coefficient, determine the Reynolds number [5,8,9].

$$Re_n = \frac{\vartheta_{sur} * d_n}{\nu} \tag{8}$$

$$Re_n = \frac{\vartheta_{bos} * d_n}{\nu} \tag{9}$$

$$Re_n = \frac{\vartheta_{dr} * d_n}{\nu} \tag{10}$$

If the Reynolds number is in the range $Re_n \leq 2300$, the coefficient of hydraulic friction is determined by the formula:

$$\lambda_n = \frac{75}{Re_n} \tag{11}$$

If the Reynolds number is in the $2300 < Re_n < 6 * 10^4$ range, the hydraulic friction coefficient is determined using the following Blasius formula

$$\lambda_n = 0{,}3164 * Re_n^{-0{,}25} \tag{12}$$

Where, $L_{B\to N}$ – distance of high pressure hose from tank to pump. $L_{B\to N}$=1 m; d_{110}– Inside diameter of high pressure hose type 110, d_{110}=101,6 mm; ρ– Density of Tellus 46 type hydraulic oil, kg/m^3; ϑ_{sur}– working fluid velocity in the suction pipeline, ϑ_{sur}=1,24 m/sec; ϑ_{bos}– working fluid velocity in the pressure pipeline, ϑ_{bos}=2 m/sec; ϑ_{dr}– fluid velocity in the flowing pipe, ϑ_{dr}=10 m/sec; $L_{N\to F}$– distance of high pressure hose from pump to high pressure filter, $L_{N\to F}$=2 m; d_{31}– inner diameter of high-pressure hose type 31, d_{31}=20 mm; $L_{F\to R}$– distance from high pressure filter to rotor spool valve 6/3, $L_{F\to R}$=3,2 m; d_{38}- inner diameter of high-pressure hose type 38, d_{38}=25 mm; $L_{R\to Gt}$– distance of high pressure hose from control valve rotor 6/3 to control valve 8/3, $L_{R\to Gt}$=3 m; $L_{Gt\to P}$– 8/3 distance of high-pressure hose from the hydraulic control valve to the excavator boom plate, $L_{Gt\to P}$=2,7 m; $L_{P\to M}$ – distance of pipe from slab to coupling, $L_{P\to M}$=1.5 m; $L_{M\to Gs}$ – distance from clutch to hydraulic arm cylinder, $L_{M\to Gs}$=2,7 m; $L_{Gs\to M}$– distance from arm cylinder to coupling, $L_{Gs\to M}$=2,7 m; $L_{M\to P}$– distance of pipe from coupling to handle plate, $L_{M\to P}$=1.5 m; $L_{P\to Gt}$ – distance from plate to spool valve 8/3, $L_{P\to Gt}$=2.7 m; $L_{Gt\to B}$ – 8/3 distance from the hydraulic spool valve to the hydraulic tank, $L_{Gt\to B}$=3.

In the following case we will also determine the local pressure losses in the $\sum \Delta P_{ul.q}$ connections, dividing them into suction, discharge and flowing pipelines, according to the formula [6]:

$$\sum \Delta P_{ul.q} = \Delta P_{sur} + \Delta P_{bos} + \Delta P_{dr} = \xi * N * \rho * \left(\frac{\vartheta_{sur}^2}{2} + \frac{\vartheta_{bos}^2}{2} + \frac{\vartheta_{dr}^2}{2}\right) MPa \tag{13}$$

Where, ξ- coefficient of local resistance in connecting joints. Suppose the average value is ξ=1.5 for all clamps [3]; N- number of connectors. N=22 pc.

Let's calculate the maximum resistance coefficients of all the hydraulic elements involved in the movement of the last arm P, based on the data given in [3].

$$\sum \Delta P_{gid.el} = \xi_F + \xi_{X.kl} + \xi_{R.gt} + \xi_{Gt} + \xi_{Tes.kl} + \xi_{Kav.kl} + \xi_{Dr.tes.kl} MPa \tag{14}$$

Where, ξ_F- local resistance coefficient of the filter, ξ_F=2; $\xi_{H.kl}$– local resistance coefficient of the safety valve, $\xi_{H.kl}$=3; $\xi_{R.gt}$– local resistance coefficient of the rotor spool valve 6/3, $\xi_{R.gt}$=5; ξ_{Gt}- local resistance coefficient of the spool valve 8/3, ξ_{Gt}=5; $\xi_{Tes.kl}$– local resistance coefficient of the check valve, $\xi_{Tes.kl}$=3; $\xi_{Kav.kl}$– local resistance coefficient of the cavitation valve, $\xi_{Kav.kl}$=3; $\xi_{Dr.tes.kl}$ – local resistance coefficient of the throttle check valve, $\xi_{Dr.tes.kl}$=3.

In a HITACHI EX-1200-6 type hydraulic excavator, the total pressure loss to the arm can be brought to the following condition by applying pressure to the power units 2 by hydraulic pumps P1 and P2.

$$\sum P_{ruk.um} = \sum P_{ruk\ P1} + \sum P_{ruk\ P1}\ MPa \tag{15}$$

So, from point 1 to point 3 when digging with the bucket and the hydraulic system when moving the hydraulic cylinder rod, calculate the pressure loss using expressions (6)-(15).

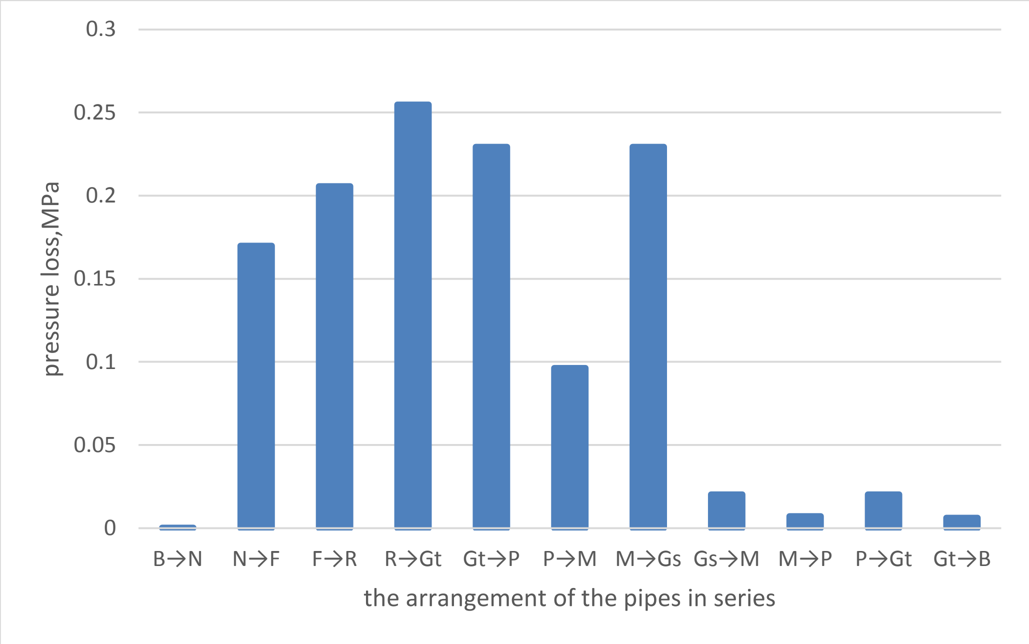


Figure 3: Pressure loss graph from point 1 to point 3 for the handle.

When performing the calculations, the result shows that each pump had a pressure loss of 2.7 MPa in the hydraulic system. At the starting pressure of 35 MPa shown in Table 1, there is a pressure input of 32.3 MPa to the crank arm hydraulic cylinder.

The head loss for the handle from point 1 to point 3 at the points where the working fluid enters the hydraulic system at head pressure, due to the displacement of the high pressure spool valve and the hydro distributor, is 62-92% of the total head (Fig.3).

3. Conclusion

So, from the results of our analysis we can see that the overall performance of a hydraulic mining excavator depends on the useful and reliable operation of the hydraulic system, the normal overall efficiency factor in the clamping sequence in the hydraulic system operation study [7,8,9,10,11,12] confirms the loss. Our study found that the hydraulic systems of a HITACHI EX-1200-6 type open-pit hydraulic excavator used in the Kyzylkum steppe zone accounts for 74-80% of the loss of normal total useful life in open pits. As a result, faults in the hydraulic system of the hydraulic excavator will have a large impact on the gauge above, and will not affect the overall gauge of the hydraulic excavator. Keeping the hydraulic system in good working order during operation is therefore relevant, and the fact that developments and research from scientists around the world exploring these solutions deserve attention and should be analyzed can ensure that they bring the best solutions.

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