Increase of durability of an amphibious vehicle water jet propulsion drive

Pavel Nenashev¹*, Sergey Abdulov¹ and Alexander Taratorkin²

¹Federal State Budgetary Educational Institution of Higher Education "Kurgan state university", 6354, Sovetskaya street, Kurgan, 640020, Russia
²Institute of Engineering Science, Ural Branch of the Russian Academy of Sciences, 34, Komsomolskaya street, Ekaterinburg, 620049, Russia

* pasteer@mail.ru

Abstract. The analysis results of dynamic loading of the amphibious vehicle’s water jet drive are presented, the hypothesis about the durability limitation of the drive due to the onset of resonance forced and parametric oscillations is advanced. Solutions of that problem are justified.

Increase of reliability of tracked amphibious vehicles is a topical problem. The divided mechanical drives with spatially arranged drivelines (cardan drives) are used in designs of transmissions of tracked amphibious vehicles. As a result of experimental studies and operation of prototypes of tracked amphibious vehicles with the similar layout of driveline, with sufficient strength margins, the limited durability of components of drivelines is revealed. Limited durability of drives’ elements become apparent in damages of shafts’ splined connections, supports and angle gearboxes’ housings, impellers, spontaneous loosening of thread connections of angle gearboxes’ fastenings, etc. The analysis of methods of design calculations demonstrates that the well-known techniques do not pay due attention to the estimation of specifics of drives dynamic loading.

This work considers theoretic and experimental research of drives dynamic loading as well as justification of ways of its decrease and an increase of drives’ durability.

The objective is achieved by dynamical and functional mathematical models of systems, simulation modeling of dynamics, oscillating processes analysis with variation of parameters that provide system dynamic stability. Mobility and maneuverability on the water are provided by water jets use. Failure of the one of elements leads to vehicle mobility disturbance on the water, in this regard special reliability requirements for these elements are imposed.

Kinematic scheme of the drive is given in Figure 1. Water jets drive with axial quadruple threaded screws of impellers is fulfilled by drivelines (cardan drives) from Left (L) and Right (R) of transmission output shafts with rotational speed 1.467 times more than rotational speed of engine . Due to the transmission layout, installation angles of drivelines on each side are different. Transmission is displaced relative vehicle centerline (Table 1).
Table 1. Values of angular coordinates that determine the spatial arrangement of drivelines

| Angle $\gamma_1$ (horizontal plane) | Left side | Right side |
|-----------------------------------|-----------|------------|
| shaft 1                           | 3°42′     | 2°28′      |
| shaft 2                           | 16°43′    | 16°43′     |

| Angle $\gamma_2$ (vertical plane) | Left side | Right side |
|----------------------------------|-----------|------------|
| shaft 1                          | 7°29′     | 4°15′      |
| shaft 2                          | 10°27′    | 10°27′     |

The gear ratio of bevel gear of angle gearbox is $u_{av} = 0.8947$. When the torque converter is interlocked the overall gear ratio of water jet drive from engine shaft to water jets impellers is $u_{dr} = 0.711$.

Interconnection of all elements, their input and output characteristics, is reflected in dynamical model consisting of mathematical models of vehicle’s transmission subassemblies including water jets drives. Such model complicates calculation algorithm significantly, however, capabilities of modern computers and software allow to develop computation program that is capable to solve differential equation systems without structural decomposition [1] of tested system.

Application of that mathematical model of dynamical processes in “engine – transmission – water jets drives – water jet” system is realized in sequential study of the specific, most indicative transient running regimes of transmission with the help of computers in the special LMS ImagineLab AMESim software package that is intended for functional simulation of various technical devices [2]. Block diagram of functional mathematical model is given in Figure 2.

Developed functional mathematical model allows to:
- determine the characteristics of dynamic loading of water jet drives’ elements used for evaluation of their durability and formation of loading spectra of relevant elements;
- determine the processes of time variations of rotational speed and accelerations of the separate elements of transmission that (processes) used for the evaluation of quality of transient running regimes.

The above listed characteristics are determined for following transient process: water jets acceleration ashore and afloat; vehicle’s water entering/leaving; reverse gear switching, etc., as well as in steady-state movement regimes at constant speed. Computational results of the time function of dynamic torques on the drive shafts of the water jets when switching on/off of the reverse gear and in steady-state regime are given in Figure 3.
Figure 2. Functional mathematical model of dynamical loading of water jets drive

Figure 3. Time function of dynamic torques on the drive shafts of the water jets when switching on/off of the reverse gear and in steady-state regime

As follows from the results of simulation, in steady-state regime, there is possibility of emergence of specific forms of oscillating processes in the system such as runouts, parametric oscillations, resonances.
The oscillation frequency corresponds “cardan” rotational speed, i.e. shaft’s double rotational speed. In process of the reverse gear switching on/off the dynamic factor increases to 1.9…3.0.

Identification of system’s separate parameters, as well as correctness of the man-made assumptions were determined during experimental studies in the process of the amphibious vehicle movement on the water. During experimental studies the rotational speed of engine shaft, dynamic torques on connecting shafts and vibration accelerations of angle gearboxes on both hull sides were recorded. Analysis of the results of experimental studies showed that dynamic factor in the reverse gear switching-on regime reaches value 1.9…2.5. In all other regimes the dynamic factor is less than 1.1…1.2 (Figure 4).

**Figure 4.** Oscillograms of dynamic torque changing on cardan shafts in the steady-state regime and in the reverse gear switching-on regime.

During experiment, in the steady-state regimes, rotational speed of the engine shaft has been changed discretely with 200 rpm interval in the range from minimum stable 800 to 2370 rpm.

Following experimental data we conclude that oscillating process has “runouts” nature. The reason of it is the summing of periodic components of the torque with near frequency. Frequencies of periodic components of two drive shafts on the one hull side in all engine speed regimes are distinct from each other by 11.7 %. Torque sensor mounted on the one shaft recorded periodic component of the torque, which is formed by other shaft. This fact indicates that in functional mathematical model there is necessity to consider mutual spatial arrangement of joints of drivelines (cardan) of water jets drives.

During experimental studies the measurements of angle gearboxes’ vibration accelerations had been performing. Peak value of angle gearboxes’ vibration accelerations was 253 m/s².

Comparison of results of experimental studies and numerical simulation coincides with sufficient accuracy at confidence probability as minimum as 95 %. Analysis of amplitude and frequency characteristics of dynamic torque in all speed range of engine as well as nature of dynamic torque changing indicates about excitation of parametric oscillations in the system that limit the elements durability.

Parametric oscillations and resonances are dangerous phenomena. They take place in wide range of disturbing frequency with exponentially increasing amplitudes of the dynamic torque. In the case of the
parametric oscillations, the drive design is subjected to dangerous cyclic loading that can lead to fatigue breakdown of the drive elements. Therefore, the main aim of the design dynamic analysis of water jet drive is determination of the boundaries of the areas of dynamic instability in order to take measures for parametric resonance elimination when reworking.

In the mechanical system the parameters formed by cardan drives with asynchronous joints periodically change. In this case, the oscillation amplitude of dynamic torque in the drive is limited and significant resistance moment when the vehicle is on the water does not allow to expand the backlash in the bevel gear of angle gearbox. Therefore when analyzing the parametric oscillations excited by drivelines the drive is considered as linear system. In given system the parametric resonances are possible [9], meaning high probability of dynamical stability loss virtually at any technically possible value of disturbing frequencies.

When water jets drives operate on land – before entering and leaving the water – energy dissipation is absent. When afloat the water jets operation is accompanied with significant energy dissipation. In this case, the amplitude of parametric oscillations is most significant, most dangerous and low-sensitive to the impact of the dissipative force. System stability can be enhanced by the introduction of oscillation absorber.

Another effective way to eliminate parametric oscillations is to decrease the modulation parameter. This is achieved by limiting of installation angle of drivelines with asynchronous joints. If the layout does not allow that, then it is necessary to introduce constant velocity joints (CVJ) for which the modulation parameter of the angular velocity is an order of magnitude less than for asynchronous ones.

During experimental studies another cause of entire drive failure was found. The consequences of loosening of angle gearbox fastenings are shown in Figure 5.

![Figure 5. Consequences of loosening of angle gearbox fastenings](image)

As a result of high dynamic loading of all elements of water jet drive the effect of vibrational loosening of threaded connections is emerging – spontaneous loosening of threaded connection. In domestic and foreign literature the least attention is paid to the problem of vibrational loosening of
threaded connections. Spontaneous loosening (self-unscrewing) is a special case of vibrational displacement effect [6], i.e. the mobility of nominally motionless parts of vehicles that emerges under vibration. The main condition of the onset of vibrational displacement effect is the presence of system asymmetry, in our case the presence of so-called force asymmetry (Figure 6).

![Figure 6. System asymmetry for vibrational displacement onset](image)

Either action of constant force T (Figure 6 option 1) or a plane inclination relative to the horizon (Figure 6 option 2) that virtually does not differ from the first option can cause force asymmetry. When transverse vibration of the plane, in both cases, the body during semi oscillation either comes off the plane or the pressing force against it decreases. As a result the force T can move a body along axis x in spite of the fact that the body remained motionless without parametrical oscillations.

We will determine the forces preventing unscrewing of threaded connections (Figure 7). The moment of forces in thread we will determine considering a nut as a slider which rising on turns of thread as on the inclined plane.

![Figure 7. Scheme of forces in threaded connection: a) moment of forces when screwing, b) moment of forces when unscrewing](image)

According to formulas [4], moment needed for threaded connection unscrewing:

$$T_{uns} = T_t + T_e,$$

where $T_{uns}$ – moment needed for connection unscrewing; $T_t$ – frictional moment in turns of thread; $T_e$ – frictional moment on the end face.

$$T_{tig} = F \frac{d_2}{2} [\tan(\varphi + \psi) + f_T \frac{D_m}{d_2}],$$

where $T_{tig}$ - torque tightening, $f_T$ – friction coefficient on the thread end face; $D_m$ – mean diameter of ring of contact of nut and bolt ends.

Preliminary tightening force acting in thread:

$$F = \frac{2T_{tig}}{d_2 \tan(\varphi + \psi) + f_T \frac{D_m}{d_2}}$$

Similar force in the connection acts when thread connection unscrewing. Based on given proposition the unscrewing moment can be determined:

$$T_{uns} = F \frac{d_2}{2} [\tan(\varphi - \psi) + f_T \frac{D_m}{d_2}]$$
It is evident that the major factors influencing on force that needed for thread connection unscrewing are the moment of preliminary tightening and frictional force in thread turns and on thread end. The moment of preliminary tightening for thread connections used for assembly should be controlled and fall within the limits specified in design documentation. It is obvious that decrease of the unscrewing moment is directly connected with change of frictional force in thread turns and on thread end.

In sources [5,6] the influence of vibration on friction coefficient is considered and is shown that the major cause for friction coefficient decrease are vibrations. Condition of absence of body slipping:

\[ fN > mA\omega^2, \]  

(5)

where \( m \) – weight; \( A \)–amplitude of oscillations of vibration displacement; \( \omega \) – angular frequency of oscillations, \( f \) – reduced friction coefficient in thread, \( N \) – force from preliminary tightening.

In case of violation of a condition (5), the effect of self-unscrewing is observed. If we will look at the right part, it becomes apparent that \( A\omega^2 \) nothing else than a peak vibration acceleration. It should be noted that in cases of action of external forces in the joint plane, unscrewing occurs also in the presence of residual force of tightening, for example, when the moment transmitted by a friction on a head is more than \( T_t \), though it is less than tightening moment. The similar result is obtained when displacement of secured parts, which can be considered as rotary movement around bolt axis. [5]

The frequency of oscillations transferred from the driveshaft to angle gearbox, providing that the rotational speed of the driveshaft is 1400-1600 rpm, varies from 46.7 Hz to 53.3 Hz.

According to design documentation, angle gearbox is fastened to the supports by М10 threaded connections. According to design documentation, tightening moment of these fastening elements should be not less than 27 Nm. On the basis of tightening moment the tightening force (3) that actuating under given tightening moment and the required unscrewing moment are calculated. Resulting tightening force is 16300 N and the required unscrewing moment (4) is 19.763 Nm.

Thus, when performing the relevant calculations the given condition (5) is not met, 2138N > 3125N, meaning that self-unscrewing effect has place in threaded connection.

In view of self-unscrewing effect of threaded connection, there can be a damage of angle gearbox housing – the ears for attachment to bracket.

When considering this case in NASTRAN software package, under condition of loosening of forward support fastening, with possibility of a displacement 0.1 mm, decrease of safety factor of gearbox housing from 2.96 to 0.98 is occurring. In case of complete loosening of forward support fastening the safety factor decreases to 0.24. Such freedom of movements contributes dramatic increase of tensions in bracket, quickly leading to damage. In this case, effective way of threaded connection locking against spontaneous self-unscrewing will be application of anaerobic sealant (Figure 8).

![Figure 8. Curves of self-unscrewing of threaded connections with different locking methods [8,10]](image-url)
This problem can be solved by application of anaerobic sealant, for example, Anakrol 202. Anaerobic sealant is intended for work in vibration environment. It is applied for fixing of threaded connections in transmission, steering units, axles and suspension system, in the engines, etc. This sealant increases the unscrewing moment by minimum 15 Nm\(^7\) at the same tightening force.

Thus, the reduced unscrewing moment will be equal to the sum of the calculated moment and the moment added by sealant, and is not less than 34.731 Nm. The new reduced angle of friction(6):

\[
\varphi = \arctan \left( \frac{2T_{\text{tor}} - F_F D_{\text{sp}}}{F d_2} \right) + \psi
\]

On the basis of the reduced unscrewing moment it is possible to calculate the reduced friction coefficient obtained after application of anaerobic sealant in thread fr fпр=0.361. In view of the new reduced friction coefficient, the condition of lack of body slipping is met, 5516>3125 N, therefore the self-unscrewing effect will be absent. This fact was confirmed during experimental study. After anaerobic sealant application the self-unscrewing effect of threaded connections of angle gearboxes fastenings was absent.

Conclusions

Reduction of dynamic loading generated by parametric resonances can be achieved by the introduction of the absorber drive as well as constant velocity joints that predetermines the durability increasing of the drive elements.

Damping of the oscillation amplitudes of the dynamic torque acting in the water jet drive that contains spatially arranged drivelines (cardans), as well as the increasing of the durability of its elements is also achieved by mutual arrangement of the driveline elements during assembling.

Elimination of onset of self-unscrewing effect, and as a result the increase of durability that reduced because of high dynamic loading of water jet drive, can be achieved by application of anaerobic sealant in threaded connections of fastenings of angle gearboxes supports for increase of unscrewing moment and elimination of decrease of tightening axial force.

References

[1] Algin V 2013 Systematization and calculation of mobile vehicle as multimass system. Dynamics of machine unit. Mechanics of vehicles, mechanisms and materials. International scientific and technical journal. Minsk Belarus v2(23) pp 5-18
[2] Chernykh I 2007 Simulation of electrical devices with MATLAB, SimPowerSystems and Simulink. 1-st edition
[3] Panovko Y 1980 Introduction to the theory of mechanical vibrations: study guide, revised 2-nd edition M.: Nauka Main office of physical and mathematical literature pp 272
[4] Birger Y and Yosilevich G 1990 Threaded and flanged connections. M.: Machinostroenie pp 367
[5] Blehman I, Blehman L, Vasilkov V, Ivanov K and Yakimova K 2012 About deterioration of equipment in vibration and shock prone environment // Bulletin of scientific and technical development No 11
[6] Blehman I 1994 Vibration mechanics. – M.:Fizmatlit pp 400. ISBN 5-02-014283-2
[7] TU 2242-003-50686066-2003. Anaerobic sealant ANACROL®-202.
[8] Ignatov A and Kechaev N 2002 Advantages of glued locking of the thread // Metizi - No, 2002.
[9] Nenashev P, Abdulov S and Taratorkin I 2018 Dynamic loading of a water jet propulsion drive of amphibious vehicles MATEC Web Conf., 224 (2018) 02042 DOI:10.1051/matecconf/201822402042
[10] Loctite. Worldwide design handbook. 1998 Loctite European Group Munich Germany pp 450