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# ШЕКТІ ЭЛЕМЕНТТЕР ӘДІСІ МЕН ТИЕУ-ЖЕТКІЗУ МАШИНАСЫНЫҢ БҰРЫЛУ МЕХАНИЗМІНІҢ ТОПСАЛЫ ТҮЙІНІНІҢ ЖҰМЫСҚА ҚАБІЛЕТТІЛІГІН МОДЕЛЬДЕУ

# МОДЕЛИРОВАНИЕ РАБОТОСПОСОБНОСТИ ШАРНИРНОГО УЗЛА МЕХАНИЗМА ПОВОРОТА ПОГРУЗОЧНО-ДОСТАВОЧНОЙ МАШИНЫ МЕТОДОМ КОНЕЧНЫХ ЭЛЕМЕНТОВ

# OPERABILITY MODELING OF THE HINGE ASSEMBLY WITH THE TURNING MECHANISM OF THE LOAD-HAUL-DUMP MACHINE BY THE FINITE ELEMENT METHOD

***Аңдатпа.****Мақалада біріктірілген рамасы бар тиеу-жеткізу машинасының бұрылу механизмінің топсалы түйінінің «саусақ-төлке» үйкеліс жұбы қарастырылады. Саусақ пен төлке материалдарының түржиынын аспаптық сәйкестендіру жүргізілді. Саусақ пен төлкенің жұмыс жағдайлары қарастырылды, олардың жұмыс беттеріндегі жүктемелердің мәні анықталды. SolidWorks бағдарламалық ортасында шекті элементтер әдісімен саусақ пен төлкенің жұмысы модельденген. Модельдеу нәтижелері бойынша жүктеме режимдерінің саусақ пен төлкенің жұмыс беттеріне әсері талданды. Бөлшектердің жұмыс қабатының бетіндегі кернеулер есептеді. Бөлшектердің түйісу аймағындағы жұмыс бетінің деформациясының шамасы бағаланды. SolidWorks ортасында модельдеу кезінде алынған зерттеу нәтижелері негізінде бөлшектердің пайдалану ресурстарын бағалау жүргізілді. Топсалы түйіннің жұмысқа қабілеттілігі мен сенімділігі туралы қорытынды жасалды.*

***Түйін сөздер:*** *Тиеу-жеткізу машиналары; бұрылу механизмі; топсалы түйін; жұмысқа қабілеттілік; модельдеу; шекті элементтер әдісі*

***Аннотация.*** *В статье рассмотрена фрикционная пара «палец-втулка» шарнирного узла механизма поворота погрузочно-доставочной машины с сочлененной рамой. Произведена приборная идентификация сортамента материалов пальца и втулки. Рассмотрены условия работы пальца и втулки, определены значения нагрузок на их рабочие поверхности. Смоделирована работа пальца и втулки методом конечных элементов в программной среде SolidWorks. По результатам моделирования выполнен анализ влияния нагрузочных режимов на* *рабочие поверхности пальца и втулки. Произведен расчет напряжений на поверхности рабочего слоядеталей. Выполнена оценка величины деформации рабочей поверхности деталей в зоне их контакта. На основе результатов исследований, полученных при моделировании в среде SolidWorks, произведена оценка эксплуатационных ресурсов деталей. Сделан вывод о работо-способности и надежности шарнирного узла.*

***Ключевые слова:*** *Погрузочно-доставочные машины; механизм поворота; шарнирный узел; работоспособность; моделирование; метод конечных элементов.*

***Abstract.*** *The article considers the friction pair "pin-bushing" of the hinge assembly of the turning mechanism of the load-haul-dump machine with an articulated frame. The assortment of pin and bushingmaterials is identified using an analytical instrument. The working conditions of the pin and bushingare considered, the values of the loads on their working surfaces are determined. The operation of the pin and bushingis modeled by the finite element method in the SolidWorks software environment. Based on the simulation results, the analysis of the influence of load conditions on the working surfaces of the finger and sleeve is carried out. The stresses in the working surface of the parts were calculated. The estimation of the amount of deformation of the working surface of the parts in the area of their contact is carried out. Based on the research results obtained during modeling in the SolidWorks environment, the operational resources of the parts were evaluated. The conclusion is made about the operability and reliability of the hinge assembly.*

***Keywords:*** *Load-haul-dumpmachine; turning mechanism; hinge assembly; operability; modeling; finite element method.*

*Introduction*. Load-haul-dump vehicles (LHD) are structurally designed according to the scheme (Figure 1) with a articulated frame [1]. The turning angle of the frame is up to 42.5 degrees [2]. The rotation mechanism folds the LHD half-frames relative to the coupling joint [3]. A hydraulic cylinder is installed on the rear half-frame of the LHD. The eyelet of the hydraulic cylinder rod is fixed to the front half-frame by means of a hinge assembly [4]. The force on the hydraulic cylinder rod is transmitted to the front half frame through a friction pair "pin-bushing". They are shown in Figure 2. This force (about 730 kN) is enough to fold the LHD half-frames during the working cycle.

**Figure 1.** Caterpillar R1300G underground loader with articulated frame

**Figure 2.** Hinge assembly’sfriction pair "pin-bushing"of the turning mechanism of the Caterpillar R1300G underground loader

Failures occurred during the operation of LHD. The hinge assemblies were dismountedindependently due to wear [5]. Technical impacts of a repair nature (to restore the efficiency of the PDM by replacing worn parts of the hinge assembly) should be attributed to corrective measures. It is necessary to evaluate the strength and wearability of the friction pair "pin-bushing" to develop technical solutions aimed at improving the operability of the hinge assembly.

Thus, the study aimed at ensuring the operability of the hinge assembly of the LHDturning mechanism is relevant and practically significant.

The object of this study is the hinge assemblies of the LHDturning mechanism.

The subject of the study is the operability of the hinge assembliesof the LHDturning mechanism.

The purpose and main idea of this scientific research is to model and analyze the operability of hinge assemblies for the development of technical solutions to ensure their higher operability.

In this regard, the objectives of the study were formulated. They include modeling the operation of hinge assemblies by the finite element method with a specialized Solid Works computer program, analysis and evaluation of their strength and wearability, as well as forecasting their resource indicators during operation.

*Literature Review*. The results of studies on the operability of LHDs and their hinge assemblies are presented in [6-8]. In [6] it was noted that the components and assemblies of the chassis of mining machines that do not have a closed (isolated) lubrication system are subject to wear due to the ingress of abrasive into them. In [7], LHD parts and assemblies that are most susceptible to wear were identified. These include elements of hydraulics (hydraulic cylinders), half-frames, booms, bellcranks and buckets (bushings, pins and fasteners). In [8] it was found that during the operation of the LHD, wear and failure of the turning hydraulic cylinder pin repeatedly occurred. It is indicated that one of the possible causes of failure may be contact compression stresses.

The use of lubricants to reduce the wear rate of friction pairs "pin-bushing" in LHD is considered in [9-11]. In [9] it is indicated that lubricants by their nature deteriorate during their operation. It is noted that the degradation of lubricants as a result of partial evaporation, oxidation and contamination is a normal phenomenon. The task of lubrication is to control the deterioration of the quality of the lubricant used during the specified operating time. In [10], the lubrication problems of the components of wheeled vehicles (dump vehicles, loaders, drilling and anchor machines) in an aggressive environment were investigated. Wheeled machines have from 15 to 40 moving parts that are subject to high shock loads and stresses. These are connections that perform pendulum movements, as well as rotational movements at low rotational speeds of up to 10 rpm. In [11] it is indicated that the wear of the sliding bearings is associated with the loss of tightness of the seals. A decrease in tightness leads to leakage of lubricant from the contact zones of the conjugated parts, to the penetration of abrasive from the environment into these zones. This leads to an increase in the coefficient of friction and the occurrence of abrasive or hydroabrasivewear. It is noted that the intensity of wear of parts depends on the concentration of abrasive in the lubricant and leakage of lubricant from the friction unit.

In [12], it is considered the problem of loosening bolted joints in mechanical structures due to a decrease in the initial axial tension and repeated external influences in the direction perpendicular to the axis. These include bolting the hinge assembly of the turning hydraulic cylinder of the LHD. The authors [12] note that an absolute estimate of self-unscrewing is necessary to predict the service life of locking devices in real machines.

*Materials and methods of research*. The results of controlled operation of LHD in the conditions of underground mines of East Kazakhstan [13, 14] showed that the hinge assemblies belong to the structural elements that limit the reliability of LHD.

Laboratory analysis of the materials of the friction pair "pin-bushing" showed the following results [15]. The finger CAT 163-5837-06 DX18H0401 (Figure 3) was made of high-quality alloy steel similar to chromium steel grade 38XA GOST 4543-71. Bushing FR 7J-9346 (Figure 4) of the eyelet of the turning cylinder rod were made of steel similar to bearing steel grade ШХ15 GOST 801-78. The method of X-ray fluorescence analysis was used to determine the brand of the pin and bushingmaterials. A portable RF analyzer of the Hitachi X-MET8000 model was used as an analytical instrument.


**Figure 3.** Pin CAT 163-5837-06 DX18H0401

**Figure 4.** Bushing 7J-9346

Analysis of failures of the hinge assembly of the LHDturning mechanism showed that the main reason was mechanical wear of the friction pair "pin-bushing" (Figure 5 and Figure 6). It was found that the bushing had a strong ovalization of the inner working surface as a result of wear [15].

**Figure 5.** Worn pinCAT 163-5837-06 DX18H0401

**Figure 6.** Wornbushing7J-9346

*Modeling of the operability of the LHD hinge assembly by the finite element method*. The geometric parameters of the Caterpillar R1300G loader are presented in [2] and shown in Figure 7.

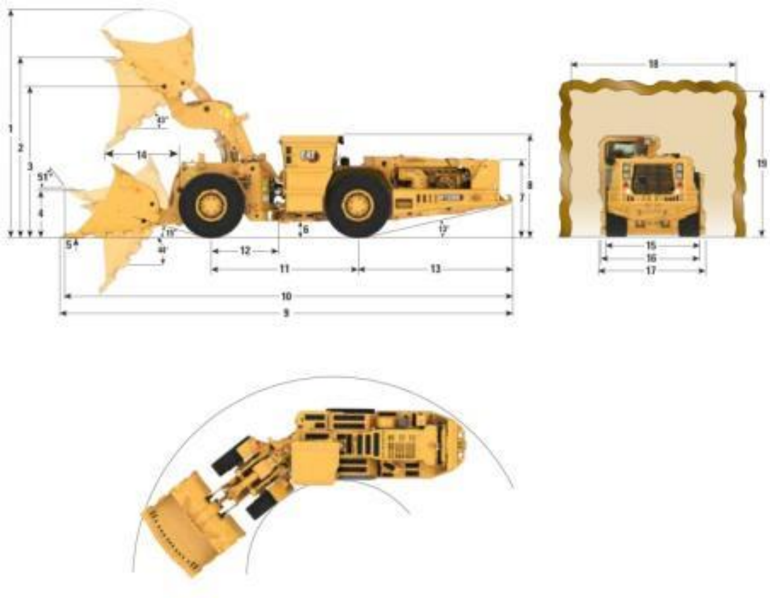


**Figure 7.** Geometric parameters of the Caterpillar R1300G loader

The design features of a LHD with a articulated frame consist in the fact that the conjugated elements of the hinge as a friction pair "pin-bushing" performs cyclic pendulum movements at 42.5 degrees. The rotation of the pin relative to the hinge axis occurs as a result of loosening of the attachment during wear in the attachment area of the pin in the bushings of the upper and lower eyelets. Linear displacement of the pin in the horizontal plane is possible with ovalization of the inner diameter of the bushing installed in the eyelet of the turning cylinder rod. There are situations when the movement of the turning cylinder rod is not accompanied by the folding of the hollow in the appropriate direction due to significant backlash in the eyelet of the turning cylinder due to the wear of the bushing.

To assess the operability of the parts of the friction pair (pin, bushing), loads were simulated, and stresses and deformations of their working surfaces were evaluated by the finite element method (FEM) in a specialized Solid Works environment. The main indicators of the Caterpillar R1300G loader and the initial values of the loads in the friction pair "pin-bushing" are presented in Table 1.

**Table1.** Initial load values for wear simulation

| Parameter | Measurement scale | Meaning |
|---|---|---|
| Operating Mass | kg | 29702 |
| Rated Payload | kg | 6800 |
| Length – Overall (Digging) | m | 8915 |
| Width – Machine with Bucket | m | 2195 |
| Length – Wheelbase | mm | 3050 |
| Length – Front Axle to Hitch | mm | 1520 |

| | | |
|---|---|---|
| Length – Rear Axle to Hitch | mm | 1530 |
| Width – Overall Tire | mm | 1900 |
| Width of Tire | mm | 444,5 |
| Track width | mm | 1455,5 |
| Articulation Angle | grad | 42,5 |
| Inner Clearance Radius | mm | 2825 |
| Diameter of Turning Cylinder | mm | 152,4 |
| Cylinderarea | m2 | 0,018241469 |
| Rod stroke length | mm | 449 |
| Compression | MPa | 40 |
| Rod Force | kN | 730 |

Modeling was carried out by the FEM using a specialized software product Solid Works. Figure 8 shows a 3D model of the friction pair "pin-bushing". Figure 9 shows a screenshot of tabular source data and modeling conditions in SolidWorks. The material properties of these parts are shown in Figure 10.

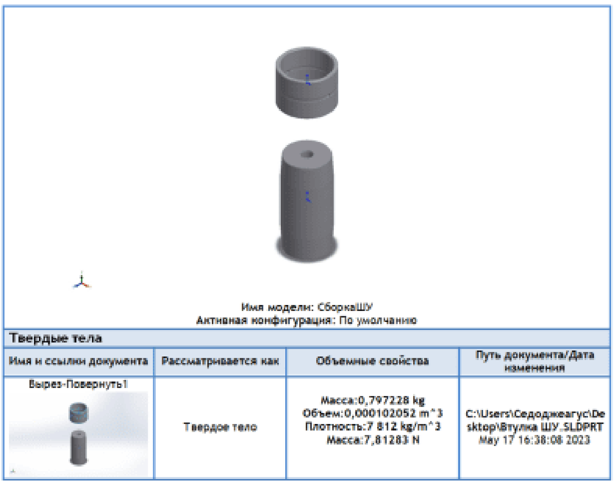



**Figure 8.** The model of the friction pair of the hinge assembly

Свойства исследования

| | |
|---|---|
| Имя исследования | Статический 1 |
| Тип анализа | Статический |
| Тип сетки | Сетка на твердом теле |
| Тепловой эффект: | Вкл |
| Термический параметр | Включить тепловые нагрузки |
| Температура при нулевом напряжении | 298 Kelvin |
| Включить эффекты давления жидкости из SOLIDWORKS Flow Simulation | Выкл |
| Тип решающей программы | Авто |
| Влияние нагрузок на собственные частоты: | Выкл |
| Мягкая пружина: | Выкл |
| Инерционная разгрузка: | Выкл |
| Несовместимые параметры связи | Авто |
| Большие перемещения | Выкл |
| Вычислить силы свободных тел | Вкл |
| Трение | Выкл |
| Использовать адаптивный метод: | Выкл |
| Папка результатов | Документ SOLIDWORKS (C:\Users\Седоджеагус\Desktop) |

**Figure 9.** Conditions for modeling a friction pair of a hinge assembly in a SolidWorks environment

Свойства материала

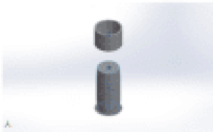
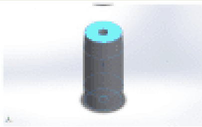
| Ссылка на модель | Свойства | Компоненты |
|---|---|---|
|  | Имя: ШХ15 ГОСТ 4543-71<br>Тип модели: Линейный Упругий Изотропный<br>Критерий прочности по умолчанию: Неизвестно<br>Предел текучести: 3,9e+08 N/m^2<br>Предел прочности при растяжении: 5,95e+08 N/m^2<br>Модуль упругости: 2,11e+11 N/m^2<br>Коэффициент Пуассона: 0,31<br>Массовая плотность: 7 812 kg/m^3<br>Модуль сдвига: 8e+10 N/m^2<br>Коэффициент теплового расширения: 1,19e-05 /Kelvin | Твердое тело 1(Вырез-Повернуть1)(Втулка ШУ-1) |
| Данные кривой:N/A | | |
|  | Имя: Сталь 38ХА ГОСТ 4543-71<br>Тип модели: Линейный Упругий Изотропный<br>Критерий прочности по умолчанию: Неизвестно<br>Предел текучести: 7,75e+08 N/m^2<br>Предел прочности при растяжении: 9,5e+08 N/m^2<br>Модуль упругости: 2,05e+11 N/m^2<br>Коэффициент Пуассона: 0,29<br>Массовая плотность: 7 900 kg/m^3<br>Модуль сдвига: 7,9e+10 N/m^2<br>Коэффициент теплового расширения: 1,2e-05 /Kelvin | Твердое тело 1(Фаска3)(Палец ШУ-1) |
| Данные кривой:N/A | | |

**Figure 10.** Properties of friction pair materials

The loads applied to the finger and the reaction forces are shown in the figure 11.

Нагрузки и крепления

| Имя крепления | Изображение крепления | Данные крепления |
|---|---|---|
| Зафиксированный-1 |  | Объекты: 2 грани<br>Тип: Зафиксированная геометрия |

Результирующие силы

| Компоненты | X | Y | Z | Результирующая |
|---|---|---|---|---|
| Сила реакции(N) | 3,30664 | -0,00267029 | -15,2178 | 15,5729 |
| Реактивный момент(N.m) | 0 | 0 | 0 | 0 |

| Имя крепления | Изображение крепления | Данные крепления |
|---|---|---|
| Зафиксированный-2 |  | Объекты: 2 грани<br>Тип: Зафиксированная геометрия |

Результирующие силы

| Компоненты | X | Y | Z | Результирующая |
|---|---|---|---|---|
| Сила реакции(N) | 3,30664 | -0,00267029 | -15,2178 | 15,5729 |
| Реактивный момент(N.m) | 0 | 0 | 0 | 0 |

**Figure 11.** Loads on the friction pair of the hinge assembly

Based on the presented initial data, it is assumed that the calculations by the FEM method will be performed when the grid is superimposed, the parameters of which are shown in the Figure 12.

Информация о сетке

| | |
|---|---|
| Тип сетки | Сетка на твердом теле |
| Используемое разбиение: | Сетка на основе смешанной кривизны |
| Точки Якобиана для сетки высокого качества | 16 Точки |
| Максимальный размер элемента | 0,00915991 m |
| Минимальный размер элемента | 0,00114805 m |
| Качество сетки | Высокая |
| Заново создать независимую сетку из неудавшихся деталей | Выкл |

Информация о сетке - Детализация

| | |
|---|---|
| Всего узлов | 21898 |
| Всего элементов | 13645 |
| Максимальное соотношение сторон | 10,893 |
| % элементов с соотношением сторон < 3 | 95,4 |
| Процент элементов с соотношением сторон > 10 | 0,132 |
| Процент искаженных элементов | 0 |
| Время для завершения сетки (hh;mm;ss): | 00:00:02 |
| Имя компьютера: | |

**Figure 12.** Информация о сетке при моделировании методом МКЭ

*Results and discussion*.The results are obtained by modeling in the SolidWorks environment (Figure 13).

Результаты исследования

| Имя | Тип | Мин | Макс |
|---|---|---|---|
| Напряжение1 | VON: Напряжение Von Mises | 0,407N/mm^2 (MPa)<br>Узел: 20595 | 1 182,506N/mm^2 (MPa)<br>Узел: 6654 |
| СборкаШУ-Статический 1-Напряжение-Напряжение1 | | | |

| Имя | Тип | Мин | Макс |
|---|---|---|---|
| Перемещение1 | URES: Результирующее перемещение | 0,000mm<br>Узел: 6354 | 0,676mm<br>Узел: 361 |
| СборкаШУ-Статический 1-Перемещение-Перемещение1 | | | |

| Имя | Тип | Мин | Макс |
|---|---|---|---|
| Деформация1 | ENERGY: Суммарнаяэнергия деформации | 0,000N.m<br>Элемент: 1620 | 0,188N.m<br>Элемент: 3549 |
| СборкаШУ-Статический 1-Деформация-Деформация1 | | | |

| Имя | Тип | Мин | Макс |
|---|---|---|---|
| Напряжение2 | VON: Напряжение Von Mises | 406 638,938N/m^2<br>Узел: 20595 | 1 182 506 112,000N/m^2<br>Узел: 6654 |

**Figure 13.** Results of simulation of the friction pair by the FEM method

Analysis of the results of modeling pin stresses by the FEM method showed that most of the load applied to the surface of the pin leads to stresses in the marginal zones of the pin(Figure 14). Under static loading, this entails the occurrence of minor energy in the middle part, and in the marginal zones of the pin, their increase takes place (Figure 15).

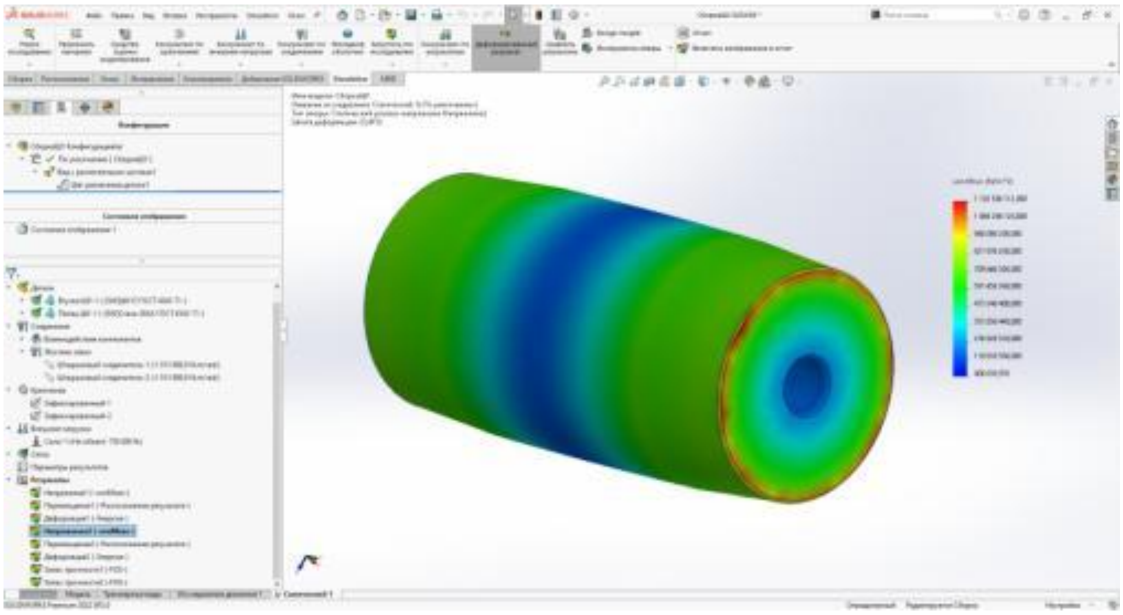


**Figure 14.** Stresses on the surface of the pin

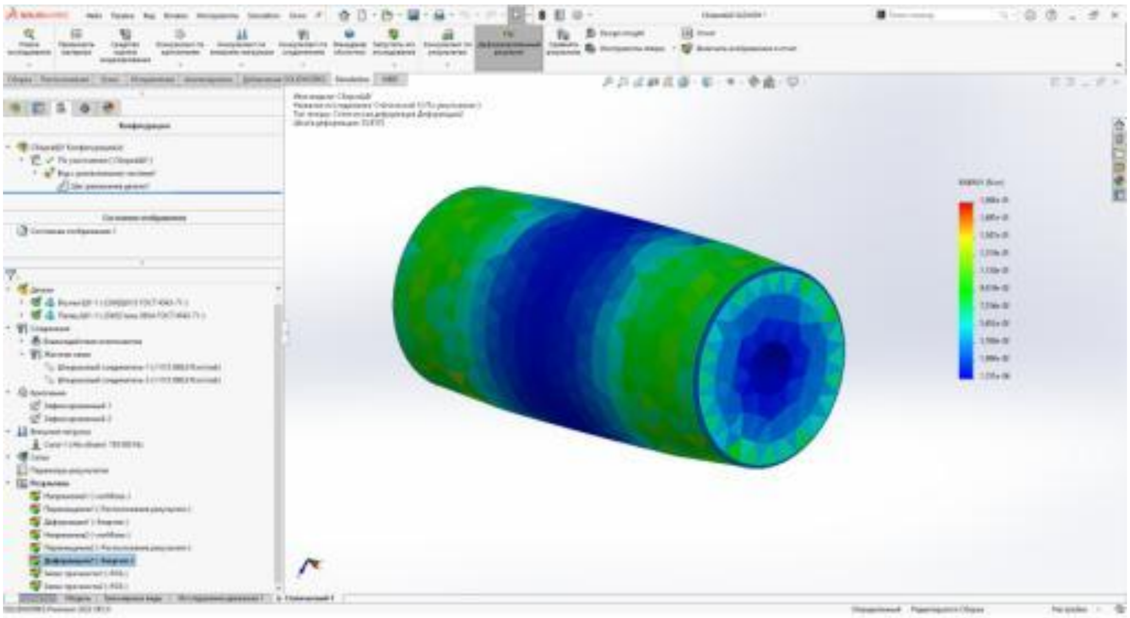


**Figure 15.** Energyofthepinsurface

Modelingthe operation of the pinconjugated with the bushing when cranking showed that in the middle working area of the pin there is an increase in the values of plastic deformation of the working surface (Figure 16). Analysis of the simulation results by the strength margin indicator showed that in the marginal zones of the pin, the strength margin of the material is not enough to ensure its operability (Figure17). This means that when the pin is loaded, it will wear out not only in the middle sliding zone in the bushing, but also in the edge zones of the pin attachment in the upper and lower eyelets of the half frame.

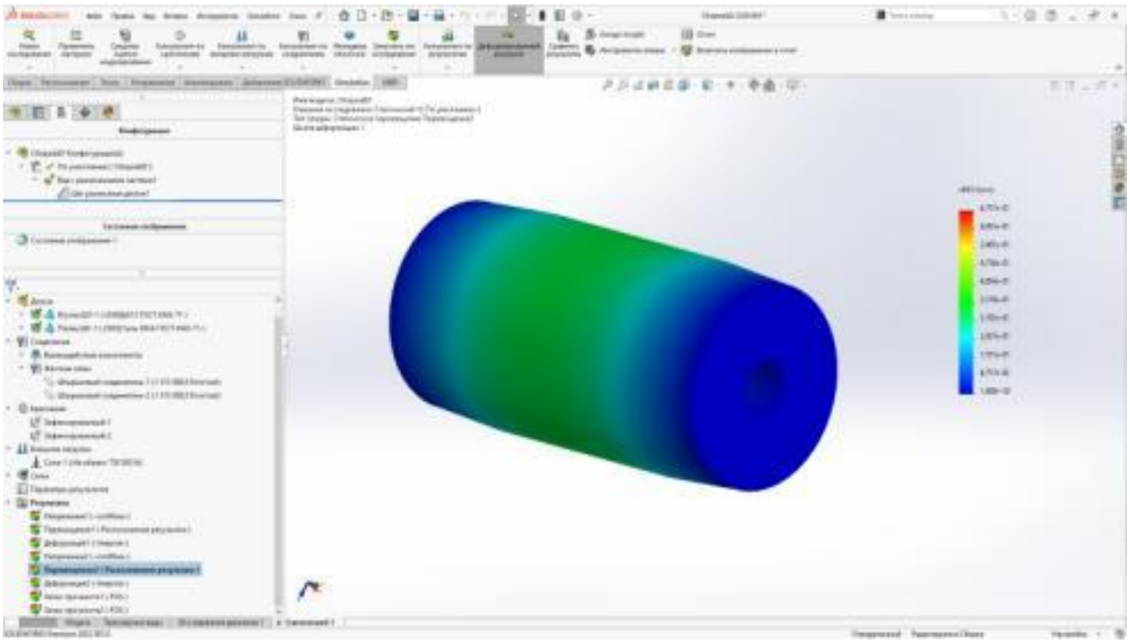


**Figure 16.** Deformation on the surface of the pin

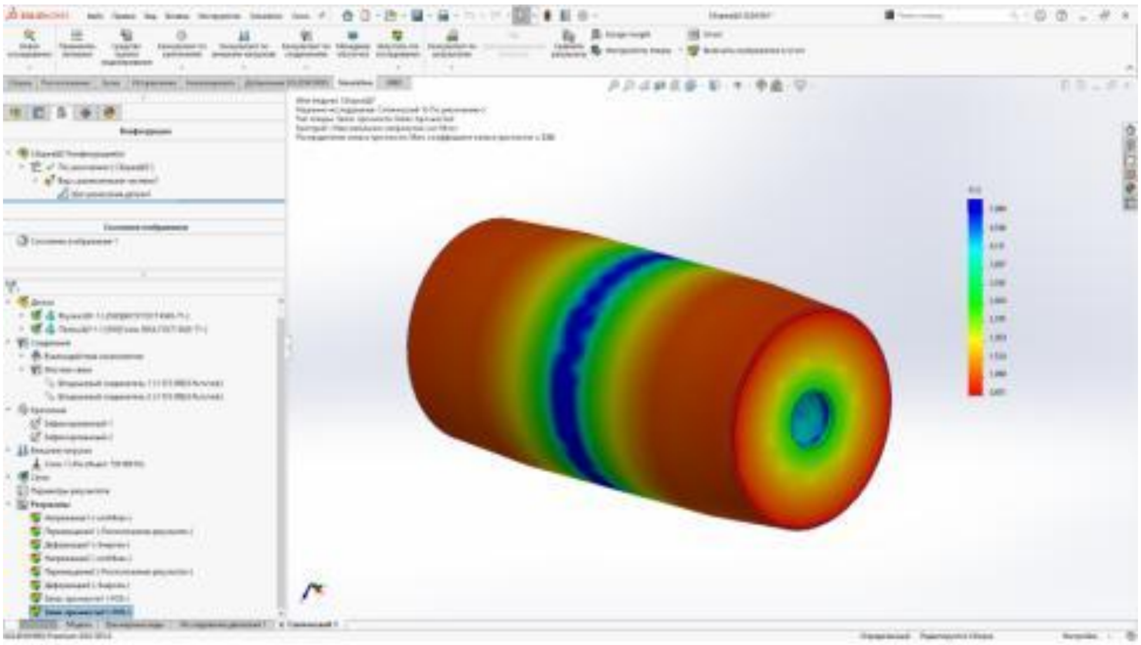


**Figure 17.** Strength margin of the material on the surface of the pin

Analysis of the results of modeling bushing stresses by the FEM method showed that most of the load applied to the bushing surface leads to stresses in the marginal zones of the inner cylindrical surface (Figure18). During static loading, foci of significant energy occur in the middle part of the inner cylindrical surface of the bushing in areas of intense load application with pendulum movement of the pin in the bushing (Figure19).

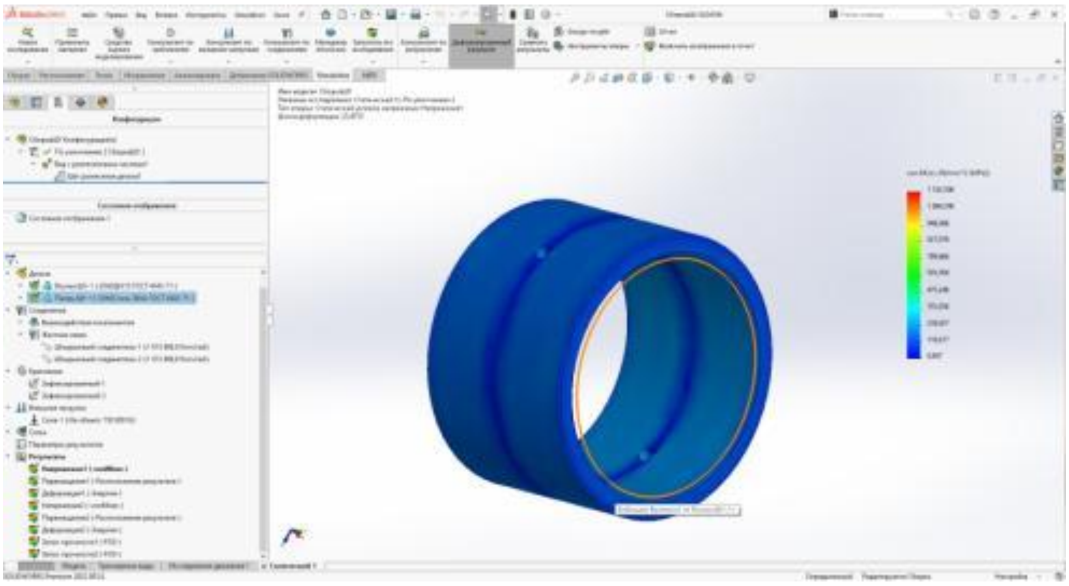


**Figure 18.** Stresses on the bushing surface

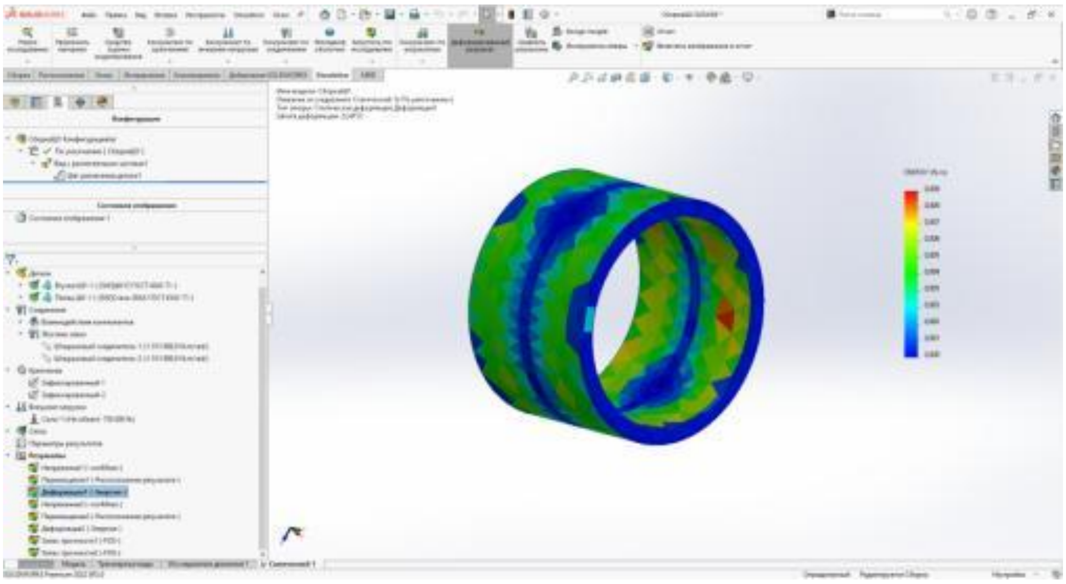


**Figure 19.** Energyofthebushingsurface

Simulation of the bushing operation conjugated with thepin when cranking showed that for the most part of the outer surface of the bushing there is a high level of deformation values of the working surface (Figure20). Analysis of the simulation results by the strength margin indicator showed that in the sliding zone on the inside working surface of the bushing, the material's strength margin is insufficient to ensure its operability (Figure21). This means that with a given load mode, the bushing will wear out intensively along the inner sliding surface.

Analysis of the simulation results of stress and wear values showed that the loss of LHD operability is reasonable, because the elements of the hinge assembly of the turning mechanism have limited strength and are prone to wear under the influence of loads.

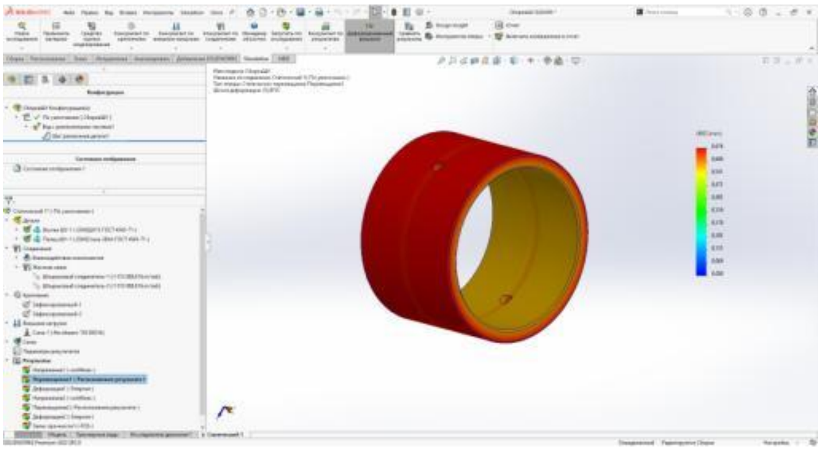


**Figure 20.** Deformation on the surface of the sleeve

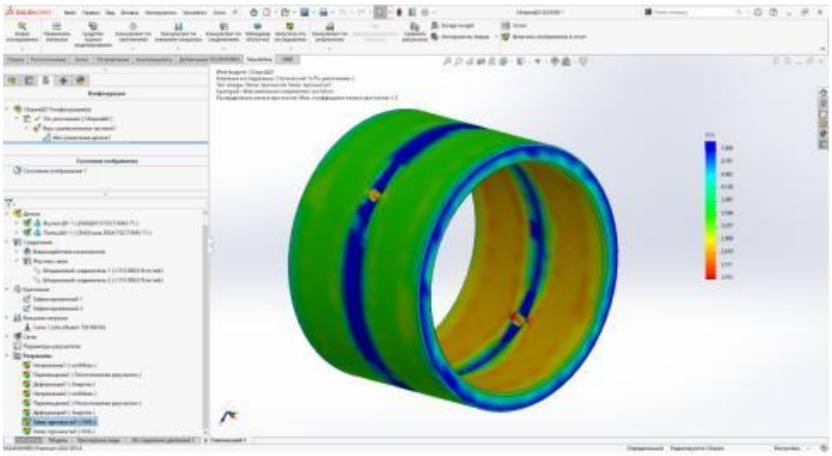


**Figure 21.** Strength margin of the material on the surface of the bushing

Thus, the results of modeling the operability of the hinge assembly parts confirm the relevance and practical significance of research and development of technical solutions to improve the operability of the hinge assemblies of underground LHD.

*Conclusions*.The article deals with the problem of ensuring the operability of articulated units of underground LHD. It is established that this problem is the result of insufficient elaboration of the design of the hinge assembly.It should be noted here, first of all, the imperfection of the design of the hinge assembly according to the criterion of its protection from the effects of abrasive and corrosive media penetrating into the working area of the friction pair "pin-bushing" of the hinge assembly and causing its wear.Secondly, the strength properties of the materials of the parts at high loads do not ensure the operability of the hinge assembly. Then, intensive wear of the hinge assembly should be considered a consequence of the design features of the LHD, load and operating conditions.It is reasonable that as a result of such wear of the parts of the friction pair, an excessive gap between them will appear.As a result, there will be additional degrees of freedom of movement of the pin and the possibility of self-dismountingof the hinge assembly.Thus, improving the design of the hinge assembly must necessarily include the development of technical solutions to ensure the controllability of the condition of the hinge assembly.

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