NUMERICAL INVESTIGATION OF A LANDING GEAR SYSTEM WITH PIN JOINTS OPERATING CLEARANCE

Wiesław Krasoń, Jerzy Małachowski, Jakub Sołtysiuk

Department of Mechanics and Applied Computer Science
Military University of Technology
Gen. Sylwestra Kaliskiego Street 2, 00-908 Warsaw, Poland
tel.: +48 22 6839654, fax: +48 22 6839355
e-mail: wkrason@wat.edu.pl, jerzy.malachowski@wat.edu.pl, kubasoltysiuk@wp.pl

Abstract

In this paper, FE method is applied to determine an operating clearance influence on the dynamics of a military transport aircraft landing gear. The numerical analysis results presentation of two-dimensional landing gear model drop tests using MSC Working Model code is also shown. Numerical results agree well with respective experimental investigation ones. The analyses using two-dimensional rigid model were performed to correlate a numerical characteristic of a shock absorber substitute model with real shock absorber characteristic. A fully deformable spatial discrete FEM model of the landing gear was developed for precise analyses aimed at determining a joint clearance influence on a considered mechanical system dynamics. Calculations were performed using the so-called direct-integration procedure, colloquially called 'the explicit integration'. Additionally at this stage, the Rayleigh damping model has been included. The non-linear dynamic analyses were performed using the LS-DYNA code. The final part presents the comparison of drop tests numerical solution results of a landing gear with assembly clearance and with operating clearance. The advantage of developed numerical method is the possibility to determine energy changing, particular components deformations, joint contact forces, what is nearly impossible to record performing experimental investigation. The presented method is applicable for a variety of boundary condition i.e. drop velocities, aircraft effective mass, etc.

Keywords: *landing gear, joints operating clearance, Finite Element Method, numerical simulation*

1. Introduction

The air transport increasing importance naturally implicates the need of research performance aimed at flight safety assurance by continuous aircraft structure reliability improvement. The necessity of landing gear reliability requirements fulfilment causes a need to apply proper research instruments at design, production and exploitation stages.

Computer simulation of a numerical model of the mechanical system (that correspond with real structure in respect of geometrical dimensions, physical features and functionality) provides a variety of possibilities to optimize and modernize considered construction by its drawbacks elimination without a prototypes building necessity. Therefore, an application of computer simulation leads to time and financial means saving.

This paper presents the creation process methodology of the landing gear spatial fully deformable model including preliminary investigation using simplified two-dimensional models. Developed and validated final FE model was used for drop tests numerical solution of landing gear with exploitation clearance to determine its influence on an investigated system performance and its operational safety.

2. Joints wear clearance

Researches concerning joint (either cylindrical or spherical) clearance issue are widely described in technical publications. It is no secret that clearance presence in mechanical systems is unavoidable. In some cases it is indispensable to assure the possibility of connected part relative motion, sometimes necessary for an assembly reason. On the other hand, clearance can appear as a consequence of imperfections, wear or deformation. This situation generates impact loads in joints that can lead to a connection life decrease, material fatigue and joint bushing fracture consequently. What is more, joint with clearance work produces noises, causes undesired energy dissipation and leads to a whole structure vibration excitation as well as its operation precision decrease. The scheme of radial clearance is shown in Fig. 1.

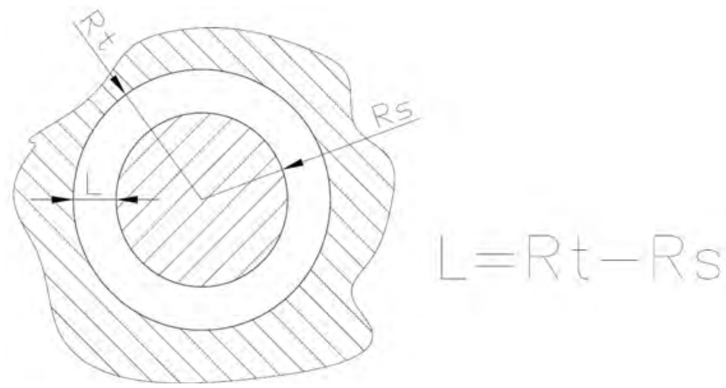

Fig. 1. Radial clearance in a pin joint

According to standard DIN 50320, wear can be defined as 'the progressive loss of material from the surface of a solid body due to mechanical action, i.e., the contact and relative motion against a solid, liquid or gaseous counter body' [1]. An ideal revolute joint introduces non-varying constraints to the system while the exploitation clearance presence enforces different approach to the joint kinematics analysis. In that case, pin receives two additional degrees of freedom – a possibility of free motion within a bushing. When it reaches the bushing the dynamics of the joint is then controlled by forces developed on the pin and bearing.

Analytical methods for dynamic analysis of mechanical systems with clearance in a joint were presented in the article [2]. Revolute joint planar models with accurate geometrical contact terms definition and a dynamic analysis of multi-body system (MBS) with revolute joint clearance were proposed. Similar mechanical system work investigation was presented in paper [7]. The results indicated that clearance increase caused reaction forces growth. This effect was illustrated in Fig. 2. Small clearance (0.0005 [mm]) practically resulted with the same reaction force history as for ideal joint. However, greater clearance value (5 [mm]) caused significant amplitudes of considered forces generating. What is more, an impact nature of these forces degraded an overall performance of considered MBS and its life.

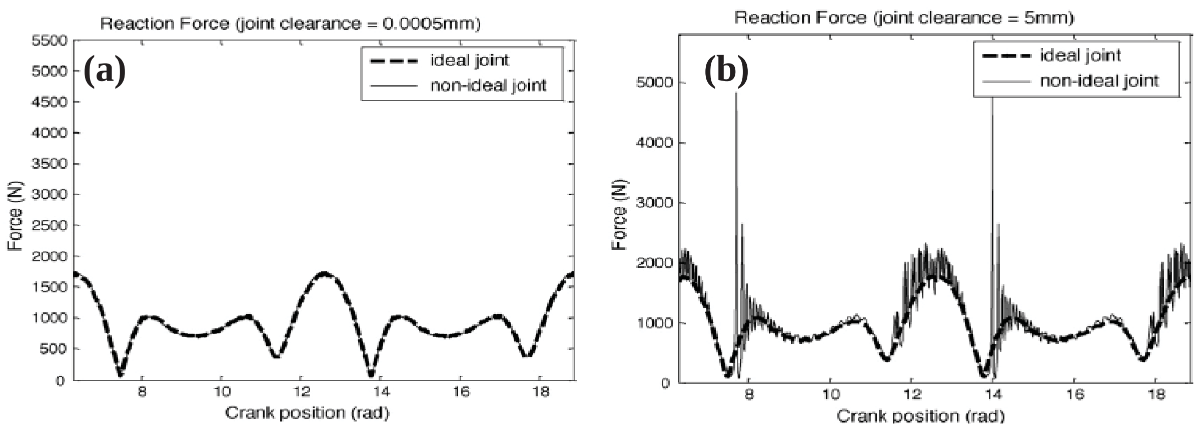

Fig. 2. The dependence of reaction force in a joint on crank position (time) [7]

3. Object of investigation

A one-wheel fixed main landing gear of polish military transport aircraft (PZL M28B Bryza – presented in Fig. 3) is an object of numerical analysis described in this paper. This light short range general-purpose high-wing monoplane is classified in the category of Short Take-Off and Landing aircrafts (STOL) and can operate on temporary prepared runways. High-wing monoplane structure protects a power transmission system. Discussed aircraft is equipped with three-wheeled fixed landing gear with front wheel controllable. A detailed description of considered object was presented in [5, 9].

The landing gear leg as its basic component is a supporting structure for an integral oleo-pneumatic shock absorber and hydraulic installation as well as stays responsible for transferring load from a wheel to the shock absorber which task is to absorb and dissipate energy of the aircraft touch-down. The action of shock absorber working medium is that liquid absorbs and dissipate energy while a compressible gas accumulate a portion of energy, as a consequence allow a damper to return to its start position and assure readiness for consecutive load absorption. The shock absorber piston rod is jointed with the rocking lever that is tied with an aircraft fuselage by a spherical joint. The investigated landing gear main components are presented in Fig. 3.

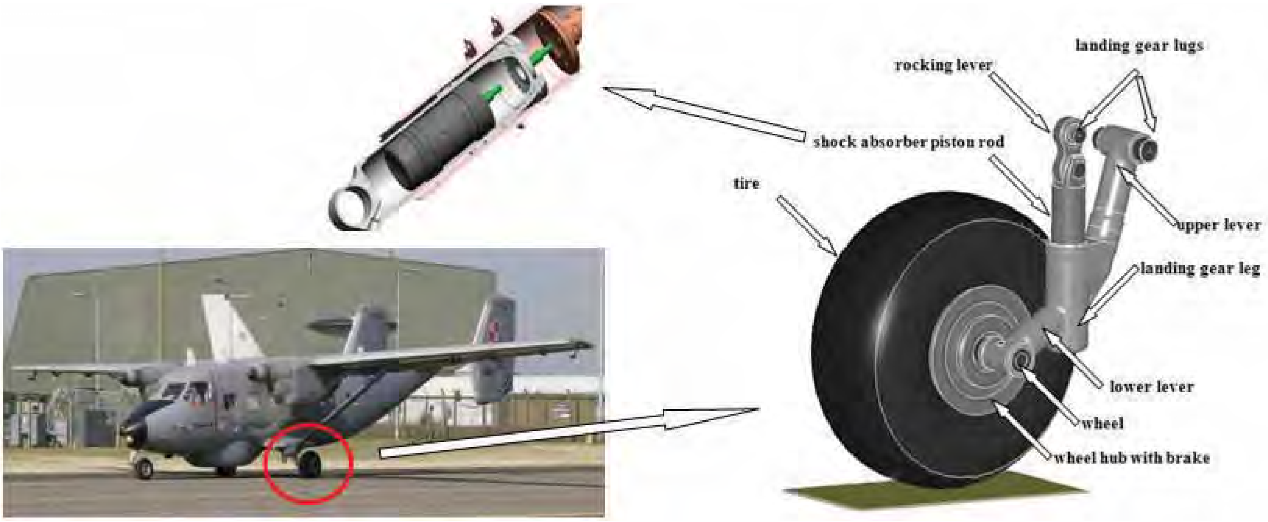

Fig. 3. PZL M28B Bryza and the investigated landing gear main components

4. A selection of shock absorber substitute characteristics for landing gear 3D model in landing simulations

An accurate representation of a damper operating can cause difficulties creating a complicated three-dimensional discrete model of a landing gear. There is a tendency to simplify modelling this component. Replacing the damper by a substitute visco-elastic element is one of the ideas that can solve this problem. The application of this element significantly simplifies shock absorber actual operating representation because it helps to avoid modelling a damper structure as well as simulating flows, liquids and gas mixing effect (Fig. 3.).

It seems to be necessary to provide a selection of characteristic constants for the substitute element in this case i.e. spring stiffness and damping constant. The main objective of performed simulation on a 2D model was to correlate the characteristic curves of resultant force at the wheel axis with experimental ones and to obtain the maximal magnitude of mentioned force (63390 N) in a drop test which refers to the ‘two-point landing’ (main landing gear touches a ground at first). The experiments on the landing gear were carried out on a test stand at the Institute of Aviation, Warsaw.

A two-dimensional model developed in MSC.Working Model (earlier studied for investigation described in paper [4]) was applied for numerical simulation. The mentioned model (Fig. 4) consists of rigid elements with shock absorber substituted by a spring with specified linear stiffness and a viscous damper. These elements parameters were adjusted during the laboratory experiments simulations.

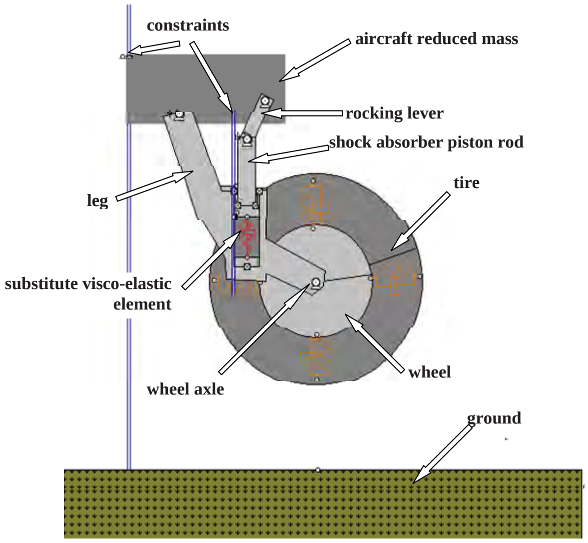

Fig. 4. Two-dimensional model developed in MSC Working Model [4]

The process of stiffness and damping coefficient selection for the substitute visco-elastic element in different landing conditions was presented in the article [8] in detail. One case was chosen for further analyses described in this paper. Its conditions are shown in Tab. 1.

The numerical test aimed at the selection of substitute element parameters was carried out in a way that guaranteed maximal value of vertical force in the wheel axle was at the same level as obtained from the laboratory experiment (63390 N). Selected parameters values are shown in Tab. 2. While a comparison of received experimental research results with computer simulation ones is presented in Fig. 5.

Tab. 1. Conditions of chosen landing case

Type:	Two-point touch-down
aircraft reduced mass that load one leg	3750 [kg]
vertical velocity	2.13 [m/s]

Tab. 2. Selected parameters for substitute visco-elastic element

spring stiffness [N/m]	1.0e6
viscous damper coefficient [Ns/m]	1.0e5

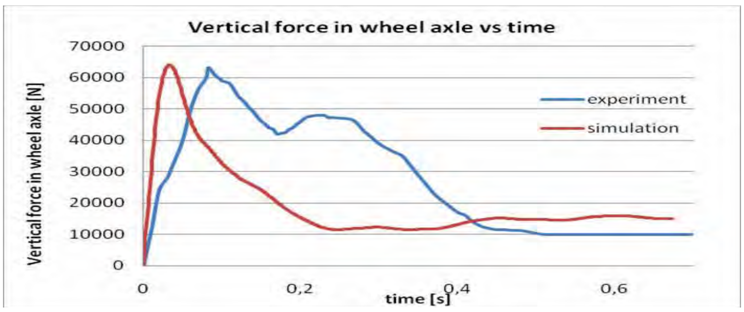

Fig. 5. Wheel axle vertical force-time diagram

5. Complete landing gear FE model

A fully deformable discrete FE model (Fig. 6.) of the considered transport aircraft landing gear was developed for precise analyses concerning its operating earlier presented in the paper [6].

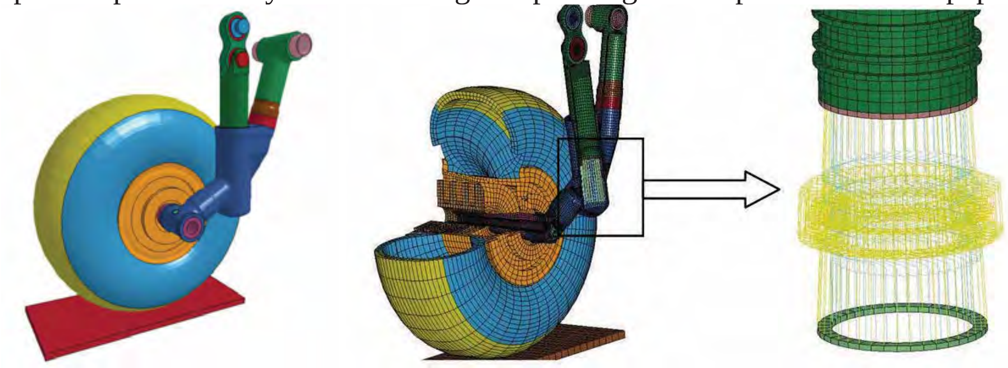

Fig. 6. Complete landing gear geometrical and FE model with the gear shock absorber discrete model

The model used in numerical simulations described in this paper contains:
- 73874 HEX8 solid elements used to describe following structural components: the upper and lower levers, the shock absorber piston rod with fasteners, the rocking lever with cup-and-ball joint assemblies – bearing races and pins, the shock absorber sleeve, the wheel axle with a fastening pin, the wheel hub with break stator and rotor discs and tire,
- 2760 QUAD4 surface elements used to determine an inner surface of the tire,
- 240 MPC elements,
- 80 discrete elements that was modelled to substitute the shock absorber operating in the considered mechanical system. The substitute model of the shock absorber consists of 40 elastic elements and 40 damping elements that were all joined directly to the corresponding nodes of additional rigid rings that were modelled between the cross-section of the bottom of the landing gear lower lever and that of the piston rod end face. The parameters (stiffness and viscous damping coefficient) for discussed elements came from previously described analyses with the two-dimensional model. Hence, equal stiffness (k_{40}) was defined for each elastic element while each damping element was characterized by equal viscous damping coefficient (c_{40}):

$$k_{40} = \frac{k}{40} = \frac{1000000}{40} = 25000\left[\frac{N}{m}\right] = 25\left[\frac{N}{mm}\right], \tag{1}$$

$$c_{40} = \frac{c}{40} = \frac{100000}{40} = 2500\left[\frac{Ns}{m}\right] = 2.5\left[\frac{Ns}{mm}\right]. \tag{2}$$

The considered substitute model of landing gear shock absorber is presented in Fig. 6.

The components of complete FE model was mainly featured with the following materials characteristics:
- 30HGSNA and 30HGSA steel for highly-loaded structures – characterize the majority of considered model components. Their detailed properties are provided in the following standards: PN-69/H-94010 and PN-72/H-84035 for the 30HGSNA steel and PN-89/H-84030 for the 30HGSA steel,
- tire material model – physical BARUMTECH tire applied in the real landing gear rubber that was featured with Mooney-Rivlin material model [3] that allowed obtaining correct results within the large displacements and a deformation range. The material properties obtained experimentally.

The aircraft reduced mass was associated with the landing gear model by adding mass elements on pins symmetry axis. The mass elements on upper lever pin symmetry axis were constrained by MPC elements to adequate nodes on the pin inner surface (Fig. 7a) while mass elements on rocking lever pin symmetry axis were equivalence with corresponding nodes of described pin (Fig. 7b).

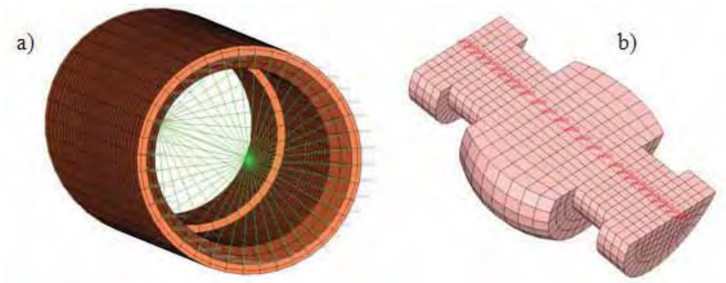

Fig. 7. Mass elements connection to upper lever pin (a) and rocking lever (b)

6. Numerical tests description

The numerical analyses were carried out to represent an instance of the aircraft touch-down and to determine the landing gear work dynamic characteristics. The conditions of chosen landing configuration are presented in Tab. 3.

Tab. 3. Conditions of chosen landing mode

Type:	Two-point touch-down
aircraft reduced mass that load one leg	3750 [kg]
vertical velocity	2130 [mm/s]
horizontal velocity	0
corresponding drop height	231 [mm]

The model was constrained at the pin joints connecting the landing gear with an aircraft fuselage. The pins were allowed to move only towards the vertical direction. These boundary conditions correspond with the real landing gear drop test circumstances [6].

The FE model of the complete landing gear was finally adopted to define a joint clearance influence on the discussed mechanical system dynamics. Developed numerical analyses allow determining energy changing, particular components deformations, joint contact forces. Those mentioned parameters are nearly impossible to record during the investigation using a physical landing gear.

A test value 0.75 [mm] of radial clearance in upper lever revolute joint was introduced (Fig. 8). The dynamic analyses were performed using LS-Dyna code for both cases: a revolute joint with assembling radial clearance (0.1 [mm]) and with exploitation radial clearance (0.75 [mm]).

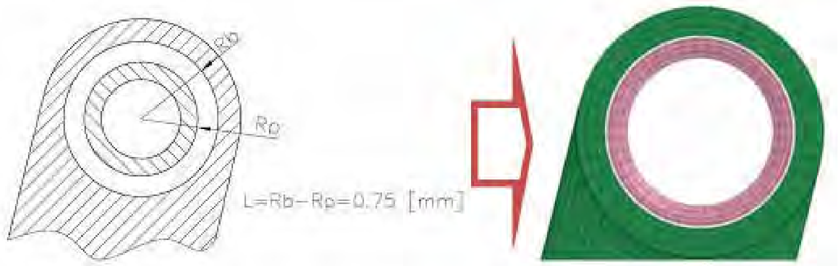

Fig. 8. Revolute joint radial clearance introduced to the landing gear model

7. Numerical tests results comparison

As a result of the carried out analyses a wide range of considered system performance parameters were obtained. A pin and bushing relative motion trajectory for the drop test of landing gear with revolute joint operating radial clearance (0.75 [mm]) is presented in Fig. 9. The maximum relative displacement of bearing and pin symmetry axles did not exceed the clearance size. The analysis of a joint resultant force-time diagram (Fig. 10) comparison for the ideal joint and the joint with the clearance leads to a conclusion that both functions are similar, however slightly higher amplitudes can be noticed for the 0.75 [mm] clearance mode. This situation directly affects von Mises stress distribution on a joint bushing (Fig. 11) – the 0.75 [mm] clearance joint operating generates about 35% higher maximum von Mises stress in a bushing (b) when compared to an ideal joint work (a). As a consequence, a joint with clearance is endangered by higher friction forces that can lead to faster joint wear degradation.

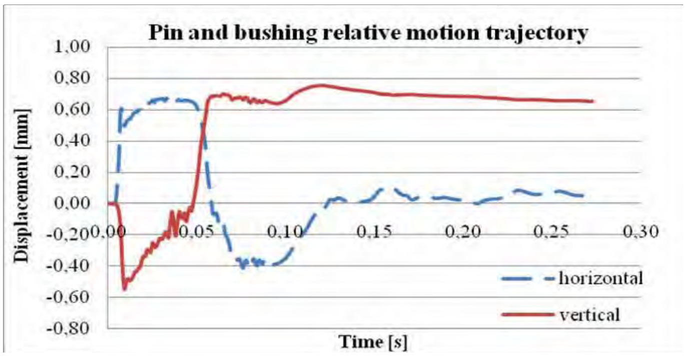

Fig. 9. Pin and bushing relative motion trajectory for 0.75 [mm] radial clearance case

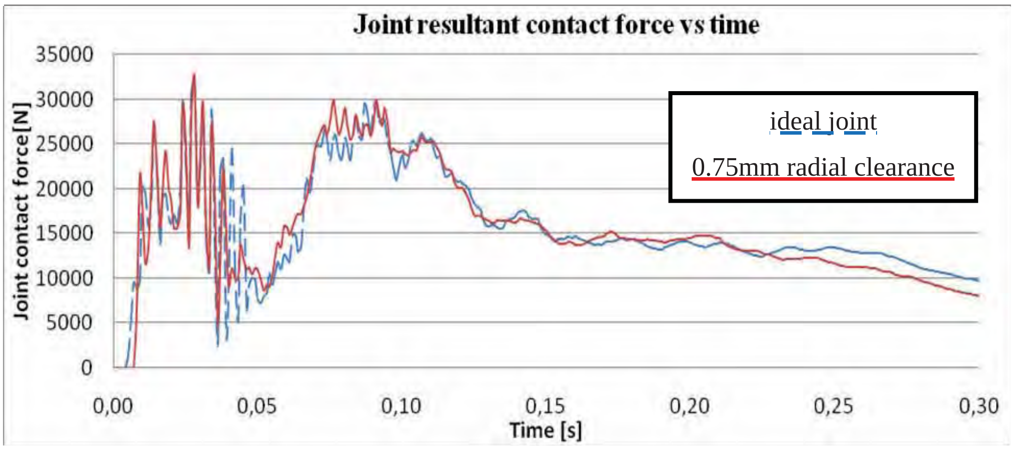

Fig. 10. Revolute joint resultant contact force-time graph

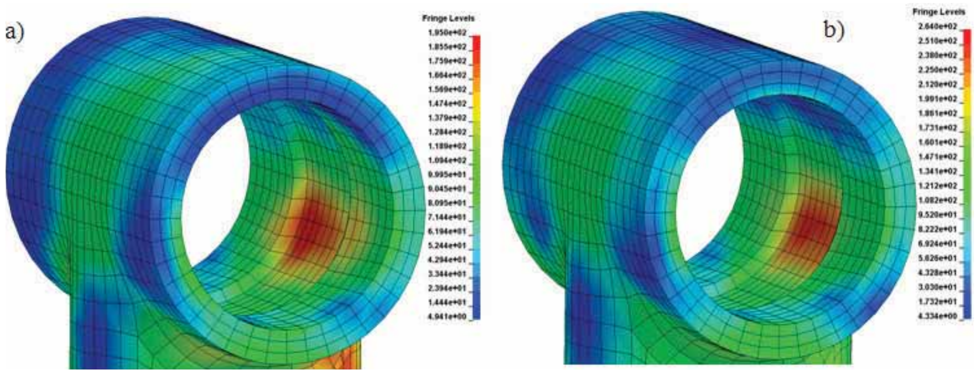

Fig. 11. von Mises stress fringes in the joint bushing zone for: (a) the ideal joint case (max 195[MPa]) and (b) 0.75[mm] clearance mode (max 264[MPa]) after 0.034[s] of drop tests time

Finally, a shock absorber deflection-time function was determined (Fig. 12). Because of a bit greater contact forces values generated in the joint area when the clearance existed, the shock absorber piston needed a slightly higher displacement to absorb the vertical drop energy than in the ideal joint case.

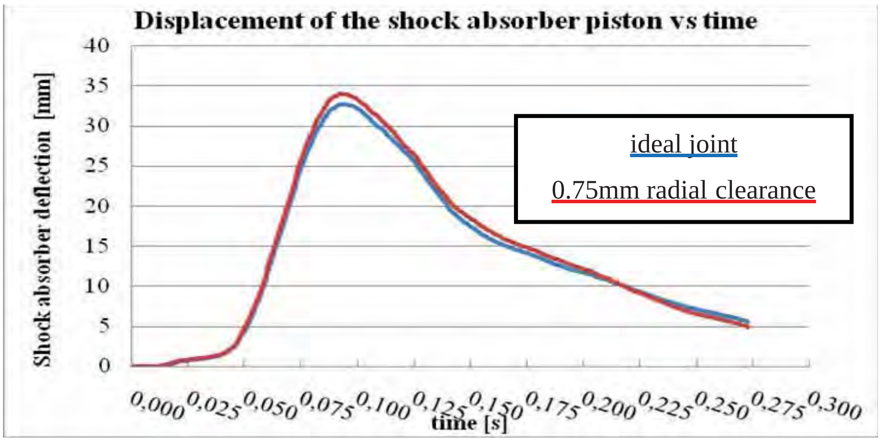

Fig. 12. Shock absorber deflection-time diagram comparison for considered cases

8. Conclusions

The analyses that were carried out revealed the difference between the landing gear operation with ideal revolute joint and joint with 0.75[mm] radial clearance. Despite the fact that the clearance existence caused the joint resultant contact force growth and as a consequence also stresses rise in the bushing, the described increases were not meaningful. This situation leads to a conclusion that the 0.75 [mm] radial clearance is not critical one. Therefore, the discussed issue require further analyses with higher clearance value to be conducted.

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