Journal of Achievements in Materials and Manufacturing Engineering
VOLUME 30
ISSUE 1
September
2008

Design and development of seal components fatigue tester

P. Czop [a], *, D. Sławik [a], G. Wszołek [b], M. Związek [b]
[a] Eastern European Engineering Center (EEEC), Control and Measuring Systems Department, Tenneco Automotive Eastern Europe, Bojkowska 59 B, 44-100 Gliwice, Poland.
[b] Institute of Engineering Processes Automation and Integrated Manufacturing Systems, Silesian University of Technology, ul. Konarskiego 18a, 44-100 Gliwice, Poland
* Corresponding author: E-mail address: piotr.czop@tenneco.com

Received 20.04.2008; published in revised form 01.09.2008

Manufacturing and processing

ABSTRACT

Purpose: The aim of the paper is research and development concerning a fatigue tester of shock absorber seal systems.
Design/methodology/approach: Analytical approach has been applied to get an understanding of a heat exchange process. The mathematical model has been formulated and validated based on the available measurements. Numerical simulation has been carried out to illustrate a heat exchange process performance.
Findings: It is possible to control a seal temperature and mechanical friction related to the rod movement into the seal at the specified velocity. The model accuracy is sufficient to perform sensitivity analysis and optimize the design.
Research limitations/implications: The components of a fatigue tester have to withstand the significant temperature differences in the range ˜ 30; +140˚C, e.g. hydraulic hoses, fittings, and pumps.
Practical implications: We combine the analytical and experimental approach to provide customized and reliable engineering solution in the area of damper component seal development. A typical seal has a lip-like design protecting the moving parts against the leakage. It can be tested in a shock absorber or externally with use of a customized test rigs. Seal tests inside a shock absorber have numerous disadvantages. A shock absorber temperature rises during longer tests and cooling phase is required. This dramatically increases tests duration performed with the use of an expensive general-purpose hydraulic testing machinery. A compressed air or water jacket is used to accelerate the cooling process. Nevertheless, there are limitations related to the physics behind the cooling process. A seal component fatigue tester allows to perform the seal component tests out of the shock absorber.
Originality/value: A new testing method provides possibility to quantify the main contributors of seal usage.
Keywords: Numerical techniques; Rapid prototyping; Engineering design; Automation engineering processes

1. Introduction

A double-tube hydraulic damper consists of a few bypassed chambers and a piston moving up and down in a liquid-filled cylinder. The piston is kinematically forced to move within the cylinder, a pressure differential is built across the piston and which forces liquid to flow through restrictions (orifices) and valves located in the piston and the base-valve assembly (Fig. 1).

The presence of the piston divides the cylinder space in to two chambers: (i) the rebound chamber, that portion of the cylinder above the piston and (ii) the chamber, that portion below the piston (Fig. 2).

The piston action transfers liquid to and from the reserve chamber, surrounding the cylinder, through the base-valve assembly located at the bottom of the compression chamber. Two types of valves are used in the shock absorber: intake valves and control valves. The intake valves are basically check valves which provide slight resistance to flow in one direction and prevents flow in the opposite direction when the differential pressure reversed. Control valves are preloaded through a valve spring to

© Copyright by International OCSCO World Press. All rights reserved. 2008

prevent opening until a specified pressure differential has built up a cross the valve. When the valve opens, the valve spring stiffness controls the amount of valve opening. Orifices of various types are used throughout the shock absorber assembly to provide flow restriction. Initial orifices, or notches in the valve seat, are used to provide a highly restricted flow path between two chambers, when the connecting valves are closed. Pre-orifices or valve restrictors are placed in series with a valve to provide a flow restriction when the valve is open.

There are two main leakage paths in a shock absorber which are sealed: leakage through the cylinder wall and the piston, and leakage between the rod and its guiding component. In the area of our interest is the former sealing system. It consists of a self-lubricated bearing, and the controlled bypass from the rebound chamber to the reserve chamber. This bypass is open when the rod moves inside the tube. The function of the bypass is dearation of the rebound chamber since the presence of entrapped air results in a large piston displacement during the oil compression stroke. The intended tester was designed to support the investigations on the rod-guide seals (Fig. 3). The main contributors of the seal durability are temperature and mechanical wear. The tester allows to perform tests under controlled exposure to internal pressure, temperature and mechanical friction resulted from a shock absorber load [1-7, 11-14, 15-22].

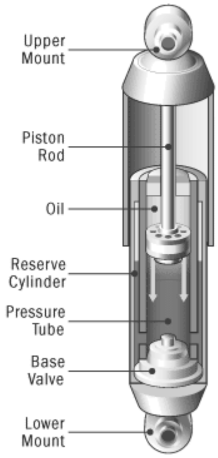

Fig. 1. Shock absorber cross-section

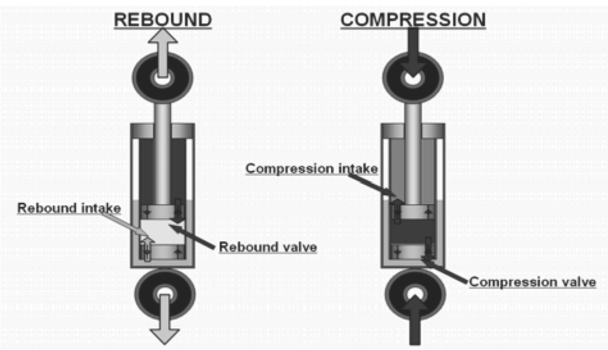

Fig. 2. Shock absorber valve system layout

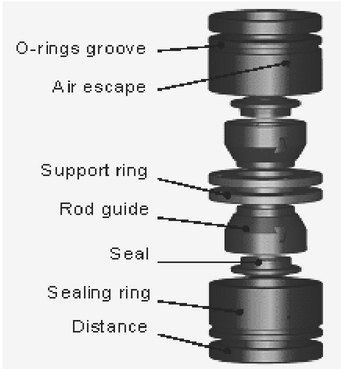

Fig. 3. Seals with rod-guide

2. Design assumptions

The hydraulic scheme of the tester is presented in Fig. 4. It consists of heating and cooling laboratory circulation devices equipped with internal reservoirs. The devices provide heating and cooling medium to the heat exchanger tube through a hydraulic circuit. The heating circulator provides precise temperature control from 5°C above ambient to 200°C and features time/temperature programming through a graphic LCD display menu. The cooling circulator provides precise temperature control from -50°C to 200°C. The temperature stability for these devices is in the range ± 0.01ºC. The hot and cold medium pumps provide the flow rate and pressure, which are required to achieve the demand heat exchange rate. Two gear pumps provide flow rate of Q=10 l/min, at pressure of p=5bar. Two on-off valves and two passive non-return valves control the flow through the heat exchanger. The additional plate cooler is used when the hot medium flowing to the cold tank. The tester has a control system based on the PLC controller and the LabView data acquisition application. The temperature is maintained with the use of an on-off controller, which is the simplest form of temperature control method. The valves and pumps are either on or off, with no middle state. [9, 10] An on-off controller switches the output only when the temperature crosses the setpoint. For heating control, the output is on when the temperature is below the setpoint, and off above setpoint. Since the temperature crosses the setpoint to change the output state, the process temperature is cycling continually, going from below setpoint to above, and back below.

To prevent damage to valves, an on-off differential, or “hysteresis,” is added to the controller operations. This differential requires that the temperature exceed setpoint by a certain amount before the output turns off or on again. On-off differential prevents the output from “chattering” or making fast, continual switches if the cycling above and below the setpoint occurs very rapidly. PID control is feasible, when the continuously operated actuators are used, i.e. a servovalve or a pump controlled via frequency changer. This solution is foreseen in the future version of the tester.

The basic component of the tester is the heat exchanger consisting of a reservoir (internal chamber) and the embedded pipe circuit feeding with the external hot/cold fluid medium.

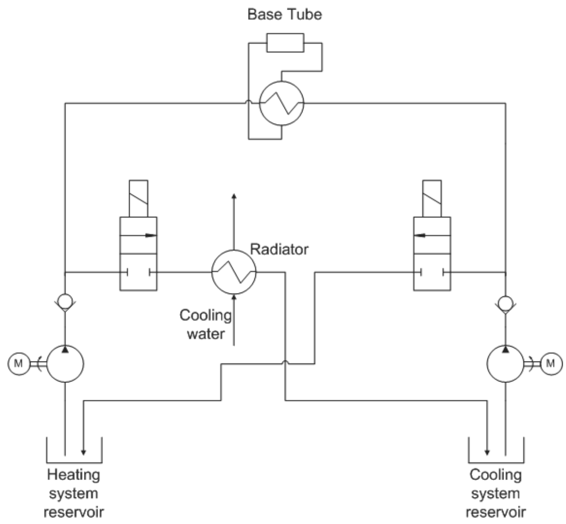

Fig. 4. Hydraulic scheme of the seal tester

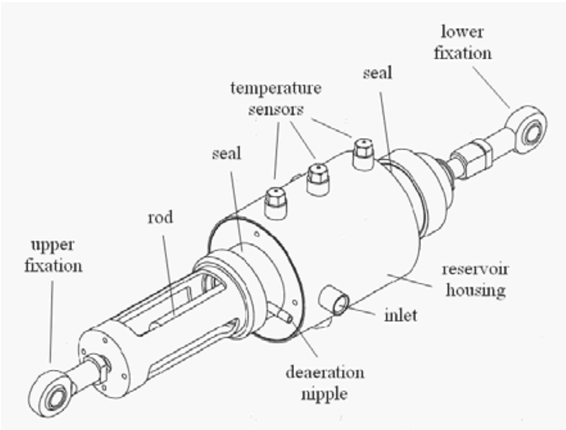

Fig. 5. Heat exchanger tube

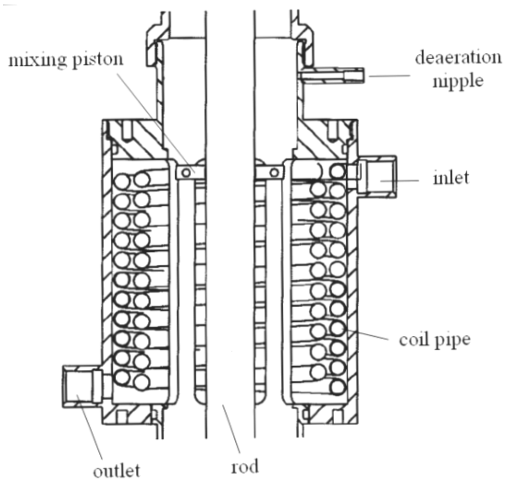

Fig. 6. Cross-section of heat exchanger tube

The reservoir is a cylindrical housing where the seals are assembled at both sides (Figs. 5, 6). The control system passes the heating or cooling medium through the on-off valves and with the use of two main pumps. The heat exchanger is mounted on the hydraulic or mechanical testing machine which provides vertical movement of the rod inside the heat exchanger (Fig. 7).

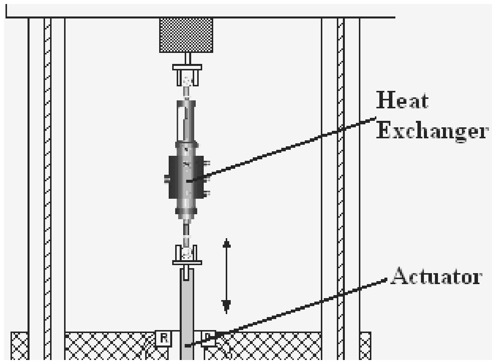

Fig. 7. Heat exchanger at the test rig

The tester has two ball joints, the top ball joint holds the whole fixture while the bottom tooling is screwed to the rod-piston post. The oil leakage can be monitored from the top through the windows in the extended nut, and from the bottom through a transparent oil pan. The oil pan is mounted on the rod body, by a metal band clip. Therefore, the oil flowing from the inside, through the seal, is gathered in the chamber. The test is continued until the oil leakage occurs. The shock absorber seal condition is quantify based on the amount of leaked oil gathered in the oil pans. This seal performance is gradually deteriorated until the leakage does not exceed the assumed amount and a test is stopped.

3. Heat exchanger model

A dynamic model was used to predict the heat exchange process between the coil pipe passing the heating/cooling energy to the internal reservoir in the heat exchanger tube (Fig. 8). The model involves two chambers corresponding to the coil pipe (chamber A) and the internal reservoir (chamber B).

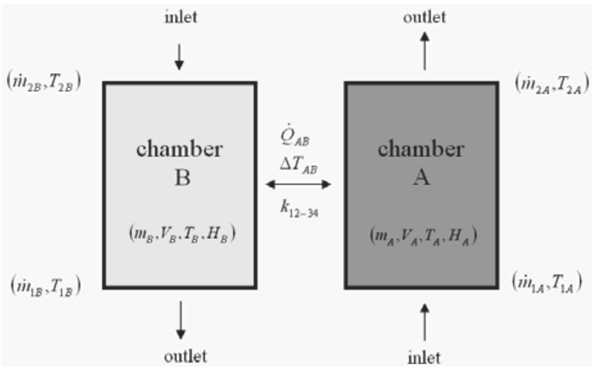

Fig. 8. Scheme of heat exchanger model

The working medium is the thermal oil flows through the coil pipe and characterized by the parameters such as the inlet temperature, pressure, mass flux, and specific heat. In results of assumed constant chamber volumes, and oil density, the outlet mass flux is equal to the inlet mass flux. Furthermore, the perfect mixing in both chambers is assumed. Therefore, the temperatures in both chambers are considered as the state variables of the model. [8] A heat exchange model defines the temperature difference ΔT_{AB} as follows:

$$\Delta T_{AB} = T_A - T_B \tag{1}$$

where

$$T_{A2} = T_A,\ T_{B1} = T_B \tag{2}$$

The outlet temperatures of both chambers are close to the average temperature in the chambers based on the assumption of the ideal fluid mixing in the tube. Therefore, the temperatures T_{A2} and T_{B1} are defined as the simulation input.

3.1. Model equations

We consider a general heat exchange system consisting of two chambers with inlets and outlets. The conservation of the mass in the chamber A yields:

$$\frac{d}{dt}(\rho_A V_A) = \dot{m}_{1A} - \dot{m}_{2A} \tag{3}$$

and for the chamber B:

$$\frac{d}{dt}(\rho_B V_B) = \dot{m}_{1B} - \dot{m}_{2B} \tag{4}$$

When the pressured inside the chamber is assumed as constant

$$\frac{dV_A}{dt} = 0 \text{ and } \frac{dV_B}{dt} = 0 \tag{5}$$

then

$$\dot{m}_{1A} = \dot{m}_{2A} = \dot{m}_A \text{ and } \dot{m}_{1B} = \dot{m}_{2B} = \dot{m}_B \tag{6}$$

The conservation of energy provides the following equations:

$$\frac{dH_A}{dt} = \dot{Q}_{1A} - \dot{Q}_{2A} - \dot{Q}_{AB} \tag{7}$$

$$\frac{dH_B}{dt} = \dot{Q}_{1B} - \dot{Q}_{2B} + \dot{Q}_{AB} \tag{8}$$

after substitution

$$\frac{d}{dt}\left(m_A c_{pA} T_A\right) = \dot{m}_A c_{pA} \Delta T_A - \alpha A \Delta T_{AB} \tag{9}$$

$$\frac{d}{dt}\left(m_B c_{pB} T_B\right) = \dot{m}_B c_{pB} \Delta T_B + \alpha A \Delta T_{AB} \tag{10}$$

where

$$\Delta T_A = T_{A1} - T_{A2};\quad \Delta T_B = T_{B1} - T_{B2} \tag{11}$$

thus

$$m_A c_{pA} \frac{dT_A}{dt} = \dot{m}_A c_{pA} \Delta T_A - \alpha A \Delta T_{AB} \tag{12}$$

$$m_B c_{pB} \frac{dT_B}{dt} = \dot{m}_B c_{pB} \Delta T_B + \alpha A \Delta T_{AB} \tag{13}$$

after rearranging

$$\frac{dT_A}{dt} = \frac{\dot{m}_A}{m_A} \Delta T_A - \frac{\alpha A}{m_A c_{pA}} \Delta T_{AB} \tag{14}$$

$$\frac{dT_B}{dt} = \frac{\dot{m}_B}{m_B} \Delta T_B + \frac{\alpha A}{m_B c_{pB}} \Delta T_{AB} \tag{15}$$

where k is the heat exchange coefficient derived in a results of analytical calculation or obtained experimentally. The tube chamber can be connected to the external source of a pressure wave. It is required to simulate the influence of pressure variability on a shock absorber. However, the flow exchange between an external wave generator and the heat exchanger tube is not foreseen. When the hydraulic port is connected to a wave generator, the flow is negligible. Under these conditions, heat exchange is assumed to be steady process characterized by the small heat exchange area equal to hydraulic fitting area. Therefore the mass flow through the chamber A is not taken into account

$$\dot{m}_{1B} = \dot{m}_{2B} = 0 \tag{16}$$

The equations governing the heat exchange process are simplified thus to the following form:

$$\frac{dT_A}{dt} = \frac{\dot{m}_A}{m_A} \Delta T_A - \frac{\alpha A}{m_A c_{pA}} \Delta T_{AB} \tag{17}$$

$$\frac{dT_B}{dt} = \frac{\alpha A}{m_B c_{pB}} \Delta T_{AB} \tag{18}$$

The equations can be formulated for the heat exchange coefficient per length of a coil pipe as follows:

$$\frac{dT_A}{dt} = \frac{\dot{m}_A}{m_A} \Delta T_A - \frac{\lambda L}{m_A c_{pA}} \Delta T_{AB} \tag{19}$$

$$\frac{dT_B}{dt} = \frac{\lambda L}{m_B c_{pB}} \Delta T_{AB} \tag{20}$$

3.2. Heat exchange coefficient

The energy flux between chambers is determined from the following equation

$$\dot{Q}_{AB} = \lambda L \Delta T_{AB} \tag{21}$$

flow coefficient k is calculated assuming that the coil pipe and reservoir fluid have the inlet and outlet parameters as presented in Table 1.

Table 1.
Inlet and outlet parameters

	Unit	Chamber A	Chamber B
Fluid density ρ	kg/m^3	900	900
Kinematic viscosity ν	m^2/s	$1.67 \cdot 10^{-5}$	$2 \cdot 10^{-5}$
Dynamic viscosity μ	Pa·s	0.015	0.015
Pressure p	Pa	0.11	
Inlet temperature t_{A1}, t_{B1}	°C	140	40
Outlet temperature t_{A2}, t_{B2}	°C	100	75
Mass flux $\dot{m}$	kg/s	-	0.373
Reynolds number Re	-	246.59	1345
Thermal conductivity α	$W/m^2 \cdot K$	129.35	113.42
Oil thermal conductivity λ_A, λ_B	W/m·K	0.117	0.125
Nusselt	-	8.851	6.237
Prandtl	-	0.282	0.047
Thermal conductivity λ	W/m·K	1.42	
Cooper conductivity λ_{HE}	W/m·K	400	400
Average heat capacity	J/kg·K	2200	1850

The procedure of the heat exchange coefficient calculation is introduced in steps in the section below. The flow velocity *w* is calculated when excitation velocity, fluid density, mass flux, and pipe area are given

$$w_A = S \cdot \sqrt{2} \cdot 2 \cdot \pi \cdot f\,, \quad w_B = \frac{\dot{m}_B}{\rho_B \cdot A_B} \tag{22}$$

The flow rate depends on the pump parameters. The Reynolds number is obtained as follows

$$\mathrm{Re}_A = \frac{w_A \cdot d_A}{\nu_A}\,, \quad \mathrm{Re}_B = \frac{w_B \cdot d_{in}}{\nu_B} \tag{23}$$

The laminar flow conditions are assumed since the Reynolds number are lower than 2300. The Prandtl number is required to determine Nusselt number:

$$\mathrm{Pr}_A = \frac{\eta_A \cdot C_{pA}}{\lambda_A} \tag{24}$$

Thus, the Nusselt number for the outer surface is as follows:

$$\mathrm{Nu}_A = 0.33 \cdot \mathrm{Re}_A^{0.6} \cdot \mathrm{Pr}_A^{0.33} \tag{25}$$

and for the inner surface is as follows:

$$\mathrm{Nu}_B = 0.023 \cdot \mathrm{Re}_B^{0.8} \cdot \mathrm{Pr}_B^{0.4} \tag{26}$$

Thermal conductivity value is determined from the formula:

$$\alpha_A = \frac{Nu_A \cdot \lambda_A}{d_A}\,, \quad \alpha_B = \frac{Nu_B \cdot \lambda_B}{d_B} \tag{27}$$

The heat transfer coefficient is determined from the formula:

$$\lambda = \frac{1}{\frac{1}{\pi \cdot d_{outer} \cdot \alpha_A} + \frac{1}{2\pi\lambda_{HE}} \cdot \ln\frac{d_{outer}}{d_{inner}} + \frac{1}{\pi \cdot d_{inner} \cdot \alpha_B}} \tag{28}$$

Where cooper thermal conductivity $\lambda_{HE} = 400$.

3.3. Simulation

The simulation has been performed to optimize the geometry and physical parameters of the heat exchanger tube to fulfill design requirements and define the temperature rising/dropping constants. A few exemplary simulation cases is presented in Figs. 9-16. The overview can be found in Table 2. The influence of flow rate in the coil pipe (case A, B, C), chamber volumes (case D, E, F), and heat exchange coefficient (case G, H) are considered in the analysis. [17, 18]

Table 2.
Simulation parameters.

Case	Flow rate [l/min]	Chamber B [liter]	Heat exchange distance L[m]
A	5	1.5	6.58
B	15	1.5	6.58
C	50	1.5	6.58
D	15	1.5	6.58
E	10	3.0	6.58
F	10	0.75	6.58
G	10	1.5	13.16
H	10	1.5	3.29

Temperature rise and drop time has been calculated in the range 80-120°C. Table 3 presents the calculated values. The measurements results correspond to the simulation case A. The error between the predicted and the measured temperature rise constant is around 20%.

4. Tester performance validation

4.1. Control system performance

The seal component tester has been validated to verify the initial design assumptions. The control and process variables are collected during the tester operation. A typical temperature operating range for tested seals is between 80°C and 120°C (Fig 17). The trial test consists of a series of 20/10 min of heating/cooling cycles, respectively. The temperature is maintained with the use of the on-off control strategy implemented in Siemens S5 controller. The temperature fluctuation is caused by the on-off control system which cycling the temperature within the assumed hysteresis limits. [15, 16] At the beginning of the cooling phase, the temperature overshoot occurs due to thermal inertia of the oil in the pipelines. This effect can be minimized with the use of optimized settings and included thermal inertia constant in the control algorithm. The problem of the lower temperature fluctuation has been solved through the improved control algorithm.

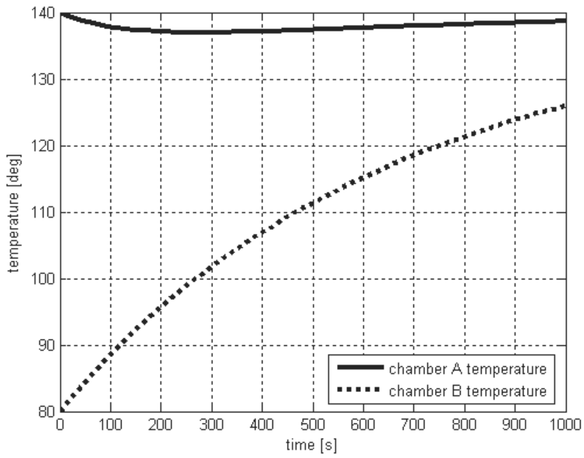

Fig. 9. Case A simulations results

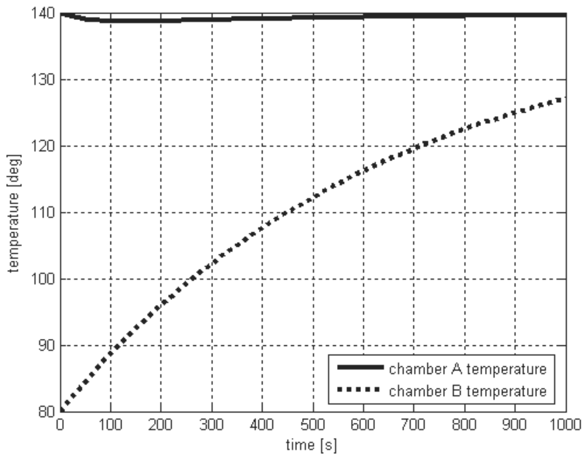

Fig. 10. Case B simulations results

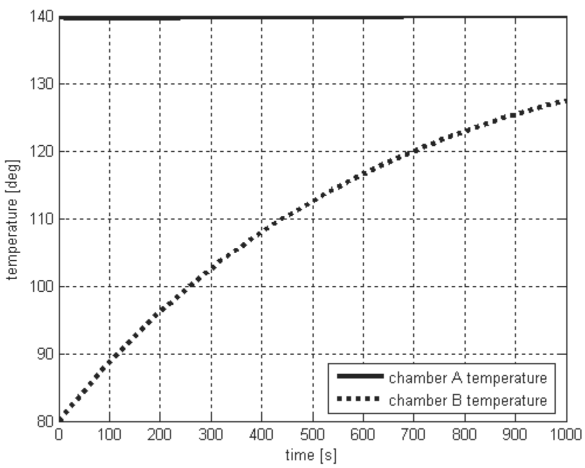

Fig. 11. Case C simulations results

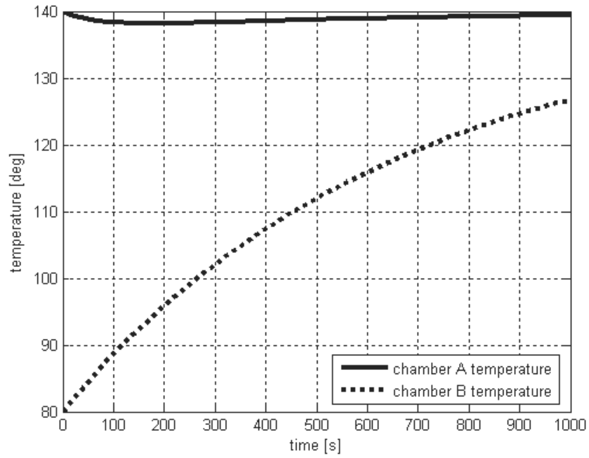

Fig. 12. Case D simulations results

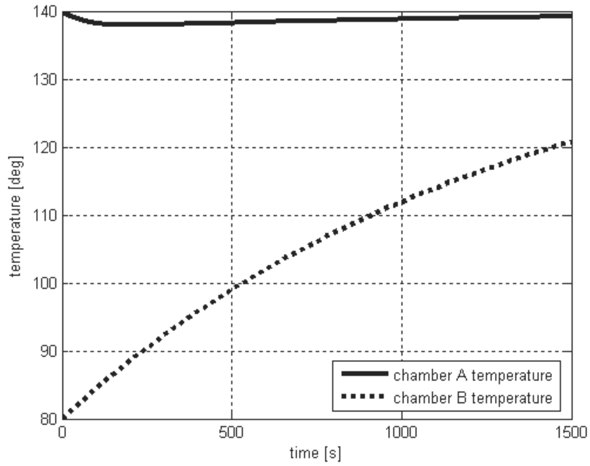

Fig. 13. Case E simulations results

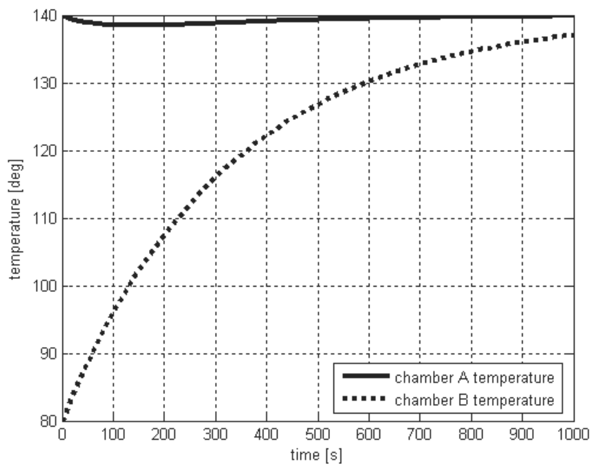

Fig. 14. Case F simulations results

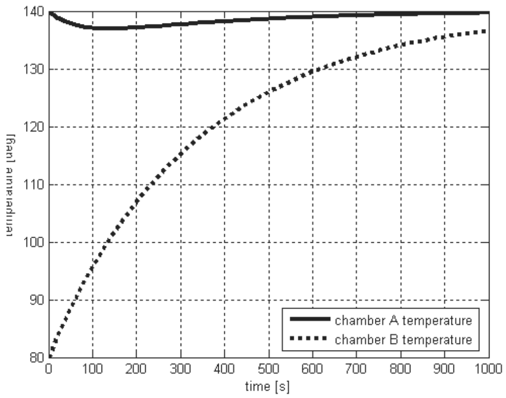

Fig. 15. Case G simulations results

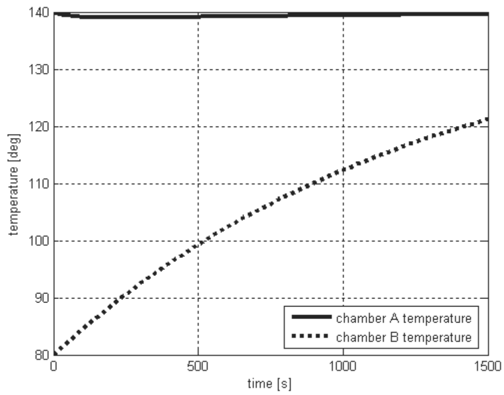

Fig. 16. Case H simulations results

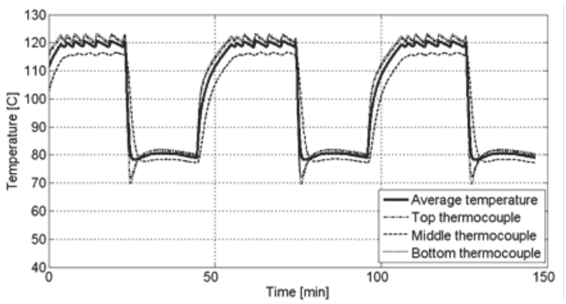

Fig. 17. Operational data in caseof the improved control algorithm

Table 3.
Rise and drop temperature constants

Case	Temperature rise constant; [ºC/min]	Temperature drop constant; [ºC/min]
Simulation A	3.26	7.27
Simulation B	3.42	7.35
Simulation C	3.47	7.50
Simulation D	3.35	7.30
Simulation E	1.65	3.52
Simulation F	6.57	14.11
Simulation G	6.40	13.33
Simulation H	1.70	3.80
Measurements	3.77	8.98

4.2. Oil/pump issue

During the initial tests of the new design it tuned out that the mineral thermal oil looses transparency and becomes polluted. Oil test indicated that it also looses its physical and chemical properties. Mineral thermal oil was replaced by a synthetic one with very similar rheological properties. No further overheating occurred, however pumps durability significantly decreased. Cold oil pump had seized after only a few cooling cycles. Both pumps were sent back to the manufacturer for checking and rebuilding. The impurities exceeded the limits according to PN-ISO 4405. We have deduced that it might have been a consequence of high value of the oil thermal absorption per area compared to the oil specification. Therefore, the tester may work with this type of oil below 100-200 hrs. Longer operation may results in passing the deposits/impurities to the valve system, pumps, and circulator tanks causing unexpected malfunctions. The oil has been changed to the synthetic thermal oil, which has comparable viscosity but creating less quantity of burnings. Unfortunately, the cold oil pump has failed due to seizure of its seals after 10 hrs of operation. It was a consequence of pure lubrication of the gear pump. In the next step, the decision to change the gear pumps into the membrane pumps has been made. This type of pump creates other obstacles, which are the maximum temperature of the used membrane, noisy and pulsation operation. On the other hand, the membrane pump does not need lubrication while it is less sensitive to changes of oil parameters with temperature, e.g. viscosity.

5. Conclusions

The seal component life tester is a device enabling a long-term durability test of damper seals. The purpose is to replace testing of a complete shock absorber by a component test which is more reliable and cost effective. The goal of the project is to design a test setup where durability test can be done only on the seal instead of on the complete shock absorber. The benefit of the new setup is increased robustness of the test (better temperature control) and decreased demand of labor (shock absorber units will no longer have to be prepared). The tooling will also enable further investigation on warranty issues thus providing means by which a demand of increased warranty periods can be addressed. The device described in this paper is constructed in a way permitting control of various test parameters. The main advantage of the approach used in the design and construction of the tester is the possibility of an independent control of the seal temperature, friction load and the pressure applied to the lip of the seal. Both the temperature and the pressure applied to the seal are controlled by varying the temperature and the pressure of the oil filling the base tube in which the seals are installed. Additionally, there is a possibility to adjust the amplitude and the frequency of the rod motion. The presented case study provides an overview of a design process of a seal component fatigue tester including the analytical considerations, the laboratory measurements, and the adequate simulation model. The simulation results have accelerated the design process reducing the costs and lead-time of the tester. Furthermore, they allow to understand the heat exchange process parameters through qualitative sensitivity analysis providing optimized design at the early development stage.

Nomenclature

- A - area [m^2]
- c_p - specific heat capacity [J/kg·K]
- d - diameter [m]
- $\dot{m}$ - mass flux [kg/s]
- m - mass [kg]
- Nu - Nusselt number
- T - temperature [ºC][K]
- p - pressure [Pa]
- Pr - Prandtl number [-]
- $\dot{Q}$ - heat flux transfer [J/s][W]
- Re - Reynolds number [-]
- w - fluid velocity [m/s]
- α - thermal conductivity per area [$W/m^2 \cdot K$]
- μ - dynamic viscosity [Pa·s]
- ρ - density [kg/m^3]
- λ - thermal conductivity per length [W/m·K]
- ν - kinematic viscosity [m^2/s]
- L - coil pipe length [m]
- H - internal chamber energy [J]
- V - chamber volume [m^3]
- d_{in} - inner diameter of the coil-pipe [m]
- S - rod movement amplitude [m]
- f - rod movement frequency [Hz]

References

[1] A.C. Atkinson, A.N. Donev, Optimum Experimental Designs, Oxford University Press, 1992.
[2] B.W. Bequette, Process Control: Modeling, Design and Simulation, Prentice Hall, Upper Saddle River, 2003.
[3] B.W. Bequette, Process Dynamics: Modeling, Analysis and Simulation, Prentice Hall, Upper Saddle River, 1998.
[4] P. Czop, D. Sławik, G. Wszołek, M. Związek, Temperature control strategy for a seal fatigue tester, Journal of Achievements in Materials and Manufacturing Engineering (in print).
[5] J. Frene, D. Nicolas, B. Deguerce, D. Berthe, M. Godet, Hydrodynamic Lubrication: Bearings and Thrust Bearings, Science B.V., Elsevier, 1997.
[6] T. Hobler, Heat transfers and heat exchangers, WNT, Warsaw, 1986 (in Polish).
[7] P. Majewski, R. Knosala, Concurrent Engineering – new look for machines constructing process, Proceedings of the 3rd International Scientific Conference "Achievements in Mechanical and Materials Engineering" AMME'94, Gliwice−Wisła, 1994, 221-228 (in Polish).
[8] J. Mrsa, B. Smoljan, Integrating quality management and cost accounting, Proceedings of the 10th International Scientific Conference "Achievements in Mechanical and Materials Engineering" AMME'2001, Gliwice−Kraków−Zakopane, 2001, 361-364.
[9] B. Mursec, F. Čus, Use of electronic catalogs for planning of machining processes, Proceedings of the 8th International Scientific Conference "Achievements in Mechanical and Materials Engineering" AMME'99, Gliwice-Rydzyna-Pawłowice-Rokosowo, 1999, 415-418.
[10] K. Psiuk, Application of the virtual screen in dialog with software user, Proceedings of the 3rd International Scientific Conference "Achievements in Mechanical and Materials Engineering" AMME'94, Gliwice−Wisła, 1994, 291-299 (in Polish).
[11] J. Świder, A. Baier, Helping of design process of the virtual creating and researching recent steering systems, Proceedings of the 11th International Scientific Conference "Achievements in Mechanical and Materials Engineering" AMME'2002, Gliwice-Zakopane, 2002, 541-544 (in Polish).
[12] J. Świder, G. Kost, J. Gorczyński, Technological processes steering with the PLC controller, Proceedings of the 11th International Scientific Conference "Achievements in Mechanical and Materials Engineering" AMME'2002, Gliwice-Zakopane, 2002, 553-560 (in Polish).
[13] J. Świder, R. Zdanowicz, Usage of the PLC controller in the model systems simulations, Proceedings of the 12th International Scientific Conference "Achievements in Mechanical and Materials Engineering" AMME'2003, Gliwice-Zakopane, 2003, 971-974 (in Polish).
[14] S. Tkaczyk, J. Michalska, Cost - quality (CQA) analysis, Proceedings of the 10th International Scientific Conference "Achievements in Mechanical and Materials Engineering" AMME'2001, Gliwice-Kraków-Zakopane, 2001, 593-596.
[15] S. Tkaczyk, M. Dudek, Usage of quality management methods in productive processes, Proceedings of the 9th International Scientific Conference "Achievements in Mechanical and Materials Engineering" AMME'2000, Gliwice-Sopot-Gdańsk, 2000, 531-534 .
[16] E. Tomasiak, Chosen questions of the steering in the technological process, Proceedings of the 3rd International Scientific Conference "Achievements in Mechanical and Materials Engineering" AMME'94, Gliwice−Wisła, 1994, 367-370 (in Polish).
[17] E. Walter, L. Pronzato, Identification of parametric models from experimental data, Springer, Berlin, 1997.
[18] D. Xue, Y.Q. Chen, D. Atherton, Linear Feedback control: Analysis and Design with MATLAB, Society for Industrial and Applied Mathematics, US, 2007.
[19] J. Świder, G. Wszołek, K. Foit, P. Michalski, S. Jendrysik, Example of the analysis of mechanical system vibrations in GRAFSIM and CATGEN software, Journal of Achievements in Materials and Manufacturing Engineering 20 (2007) 391-394.
[20] C. Cempel, Mechanical Vibrations. Introduction. Poznan University Publishing Company, Poznań, 1984 (in Polish).
[21] J. Świder, G. Wszołek, Graphs Application in Computer Analysis of Mechanical Systems. Monograph, Silesian University Publishing Company, Gliwice, 2002 (in Polish).
[22] G. Wszołek, Vibration analysis of the excavator model in GRAFSIM program on the basis of a block diagram method, Journal of Materials Processing Technology 157-158 (2004) 268-273.