An optimization-based design approach for a novel self-adjuster using shear thickening fluid

P. Nagy-György¹ · J. G. Bene¹ · C. J. Hős¹

Received: 12 April 2021 / Revised: 14 June 2021 / Accepted: 2 August 2021 / Published online: 6 September 2021
© The Author(s) 2021

Abstract

Recently, the increasingly strict safety and emission regulations in the automotive industry drove the interest towards automatic length compensating devices, e.g., hydraulic lash adjusters (lower emission) and slack adjuster in brake systems (faster brake response). These devices have two crucial requirements: (a) be stiff during high load, while (b) be flexible in the released state to compensate for environmental effects such as wear and temperature difference. This study aims to use the advantageous properties of shear thickening fluids to develop a less complicated, cost-efficient design. The proposed design is modeled by a system of ordinary differential equations in which the effect of the non-Newtonian fluid flow is taken into account with a novel, simplified, semi-analytical flow rate-pressure drop relationship suitable for handling arbitrary rheology. The adjuster's dimensions are determined with a multi-objective genetic algorithm based on the coupled solid-fluid mechanical model for six different shear thickening rheologies. The accuracy of the simplified flow model is verified by means of steady-state and transient CFD simulations for the optimal candidates. We have found that the dominating parameters of such devices are (a) the shear thickening region of the fluid rheology and (b) the gap sizes, while the piston diameters and the zero viscosity or the critical shear rate of the fluid have less effect. Based on the results, we give guidelines to design similar-length compensating devices.

Keywords Multi-objective genetic algorithm · Optimal design · CFD shear thickening fluid · Non-Newtonian fluid · Self-adjuster

1 Introduction

1.1 Motivation

Heavy-duty vehicles include a number of wear parts (e.g., brake pads, clutch disks) whose wear must be compensated for the vehicle's whole lifetime. This work seeks for a solution to compensate the wear in pneumatic clutch actuators to increase the automatic transmission system's reliability and performance.

The proper adjustment of transmission systems are currently essential for driver comfort, environmental protection, and autonomous driving (Ngo and Colin Navarrete

2013). The appropriate transmission system can adapt the gear shifting to the actual vehicle load and road conditions, which cause significant fuel savings. However, to achieve the optimal settings, the gear shifting process of the vehicles must be fully automated. In heavy vehicles, the traditional manual clutch systems are nowadays automated using an electro-pneumatic actuator to substitute manual force.

In the systems described above, the clutch actuator is responsible for reaching the time-varying required position of the clutch disk with high speed and accuracy, ensuring a good start and shifting even in extreme situations (e.g., slippery ground, steep incline). Most of these clutch actuators operate electro-pneumatically. The actuator should satisfy the speed and accuracy requirements throughout the vehicle's life span, but this is a challenging problem because the actuator piston moves within the full stroke of the cylinder as a function of the clutch disk wear. In practice, this results in a large dead volume at the beginning or end of the lifecycle. If the pneumatic cylinder has a larger dead volume, the pressure builds up slowly, and the speed decreases significantly.

Responsible Editor: Seonho Cho

✉ P. Nagy-György
pnagy-gyorgy@hds.bme.hu

¹ Department of Hydrodynamic Systems, Budapest University of Technology and Economics, Budapest, Hungary

Indeed, in Szimandl (2015) and Rad and Hancu (2017) the authors show that the dead volume has the most critical effect on the actuator performance.

Due to the reasons presented above, several attempts were made to compensate for the wear and keep the actuator piston's initial position in the same value. The active solutions use sensors and solenoids (Bader and Graf 2006), but using electronic devices and developing the proper control lead to an elaborate and expensive design. Passive solutions operate mechanically. If there is a load on the system, it activates a friction-based self-locking mechanism that restrains the self-adjuster's length change. When the adjuster is unloaded, the parts of the adjuster can move relative to each other (for more details, see Solberg (2011), Ljøsne (2015)). However, the sufficient friction coefficient between the parts can be obtained only by high-precision manufacturing, leading again to an expensive design.

Our concept takes advantage of shear thickening fluids' property; at low shear rates they behave as fluids allowing wear compensation, while at high loads they become stiff (actuation). Employing this behavior allows less strict tolerance values and hence cheaper manufacturing and the complete lack of active elements.

1.2 Literature review

Length adjusting mechanisms are essential for a wide range of application areas in the automotive industry, e.g., valve lash adjusters in engines (Nunney 2007), automatic slack adjusters in brakes (Bennett 2015), and flexible pistons in variable compression ratio engines (Shaik et al. 2007). These are mostly passive elements to enhance reliability by eliminating active control systems. Hydraulic lash adjusters were developed in Ferreira et al. (2016), Leighton and James (2018) to compensate for the effect of thermal expansion in engine control valves. The length adjusting property is based on the flow through a check valve between two hydraulic oil-filled chambers and, despite its simple layout, the design ensures optimal valve timing and hence the satisfaction of increasing requirements for noise, emissions and economy. Reaching the desired performance is still a challenging problem, which usually requires modern modeling and optimization tools (Gecim and Raghavan 2009; Okarmus et al. 2008; Sano et al. 2013).

Automatic slack adjusters aim to compensate for brake disk wear to maintain constant brake stroke and improve the reliability and performance of air brakes (Pradeep 2008; Prado et al. 2017). The operation principle relies on simple mechanical phenomena, mostly friction (Shellhause 1987) or ratchet mechanism (Prado et al. 2017). Similarly, mechanical adjusters are applied in rail brake systems as well.

The increasing demand on emissions and fuel consumption drove the interest toward variable compression ratio

engines (Ozcan and Yamin 2008; Muralidharan and Vasudevan 2011) in which the length of the piston is varied using a control system to obtain a more efficient engine.

Such mechanisms must fulfill two requirements: (a) be stiff during high load (closing valve in the case of hydraulic lash adjusters, braking in the case of automatic slack adjusters) and (b) compensate environmental changes (e.g., thermal expansion, wear) during released state (i.e., keeping the valve open in the case of a hydraulic lash adjuster and releasing the brake in the case of automatic slack adjusters). The most crucial drawbacks of these adjusting mechanisms are the complex design, including a large number of parts. Moreover, most of them cannot compensate in both directions or only at the cost of even more intricate design. Our proposed adjuster desires to provide similar features with a less elaborate layout exploiting the advantage of non-Newtonian property.

Typical extensible and compressible elements are the shock absorbers. The operating principle of these devices is that due to the external forces, pressure difference builds up between two fluid-filled inner chambers connected by (usually small) gaps, and the fluid starts to flow through the gaps, while the damper compresses or elongates. Traditionally, Newtonian (e.g., hydraulic oil, water) fluids are employed (see Dixon 2008); however, with increasing vibration isolation demands, non-Newtonian fluids became more and more significant (Tudon-Martinez et al. 2019; Yeh et al. 2014). Recent studies usually apply shear thickening fluids (Lin et al. 2018; Wei et al. 2019, 2018; Zhou et al. 2016; Zhao et al. 2018; Zhang et al. 2008; Hou 2008), frequently in the application of seismic protection (Wei et al. 2019, 2018, 2018). These dampers have progressive damping characteristics, i.e., the damping force increases faster than linearly with the deformation speed.

Shear thickening fluids (STF) are typically suspensions, i.e., solid nanoparticles [e.g., fumed silica (Yeh et al. 2014; Zhao et al. 2018; Zhang et al. 2008; Soutrenon and Michaud 2014; Tian and Nakano 2017), cornstarch (Jiang et al. 2017)] solved in a carrier fluid [e.g., silicone oil (Zhang et al. 2014), ethylene-glycol (Zhou et al. 2016; Zhang et al. 2008; Tian and Nakano 2017), polyethylene-glycol (Zhao et al. 2018; Soutrenon and Michaud 2014; Tian and Nakano 2017), polypropylene-glycol (Yeh et al. 2014; Chang et al. 2011), water (Jiang et al. 2017; Nenno and Wetzel 2014)]. Rheological properties of such fluids are complex (see Fig. 6), and in most cases, there is no simple model to describe their behavior. In most cases, the fluid has viscoplastic behavior by low shear rates, i.e., the fluid starts to flow only in the case of shear stress values above the yield stress. Therefore, the fluid has a slight shear thinning behavior in this first region, i.e., the fluid's apparent viscosity decreases with increasing shear rate or shear stress. Practically, the fluid behaves as a liquid, and it can flow easily (Region 1 on Fig. 6). Near the critical shear

rate, hydroclusters are formed, and the so-called jamming state appears. The apparent viscosity of the fluid increases abruptly (Region 2 in Fig. 6) until it exhibits a solid-like structure. After this second yield point, the mixture behaves as a deformable solid-like body. For example, a person can run across the surface of a pool filled with a cornstarch-water suspension, but if he stops, he sinks immediately (Brown and Jaeger 2014). More details about the shear thickening effect can be found in Brown and Jaeger (2014).

Although there is a wide range of application areas of shear thickening fluids, e.g., protective equipment (Gürgen et al. 2017; Mawkhlieng et al. 2020), vibration isolation (Galindo-Rosales 2016), impact resistance (Srivastava et al. 2012), up to the authors' best knowledge, there is no existing publication on shear thickening fluid-based compensator devices. The device proposed here should have similar behavior as STFs, i.e., the device should be solid-like stiff in the case of high compressible load (shear stress), while it should be flexible in the case of low load (by low shear stress).

1.3 Layout

The rest of the paper is organized as follows. Section 2 describes the idea of the STF-based adjuster, including industrial requirements and test specifications. Section 3 summarizes the applied mathematical tools like governing equations, CFD modeling, and multi-objective genetic algorithm. The adjuster's behavior is described by a system of ordinary differential equations (ODEs), in which the fluid mechanical forces are calculated by the authors' previously proposed method (Nagy-György and Hős 2021, 2019). The validity of that previously defined method is checked by means of CFD simulations for the relevant fluid rheologies in Sect. 3.3. Using these ODE models' results, multi-objective optimization is performed to find the most appropriate fluid and the corresponding geometrical dimensions of the device. The genetic algorithm's details, especially the design variable range and constraints, are described in Sects. 3.5 and 3.6. The optimization process provides the Pareto fronts and the optimized geometry for six different fluid rheology. The feasibility and dynamic behavior of these optimal designs (lying on the Pareto front) are checked by a more accurate, unsteady CFD technique. The differences between the designs and the theory behind them are detailed in Sect. 4. Finally, Sect. 5 concludes the study.

2 Design and test specification

2.1 Traditional length compensating devices

Our starting point is the STF-based damper solutions available in the literature. The traditional (so-called) monotube

dampers can be classified according to two aspects: the piston type is single-ended [Fig. 1c (Hong et al. 2008; Esteki et al. 2014), d (Snyder et al. 2001; Fali et al. 2019)] or double-ended [Fig. 1a (Yeh et al. 2014; Wei et al. 2019; Zhou et al. 2016; Zhao et al. 2018; Hou 2008; Jiao et al. 2017), b (Kim et al. 2016)] and the shape of the flow restriction [annular gap (Fig. 1a, c) or cylindrical hole (Fig. 1b, d)]. All of these consist of a cylinder, a piston, sealing elements, and the hydraulic liquid, see Fig. 1. In the double-ended rod design, the cross-section of the chambers equals each other; therefore, no compensation element is needed. If the piston moves, the volume change of one side equals to the other side's (Fig. 1a, b).

In contrast, in the case of the single-ended version, the volume changes are not equal, which means that a compensation element is needed. That is usually an accumulator cavity with an elastic membrane filled with compressed gas, mostly air or nitrogen. These elements behave as a secondary piston pushed by an air spring, symbolized in Fig. 1 with a secondary piston.

Our concept can be seen in inset (e) in Fig. 1, which is an improved version of the single-ended annular gap damper (Fig. 1c). The compensating piston is moved to the upper chamber bordered by the piston, and an additional limiting element is built-in. The former modification is necessary

Fig. 1 Typical damper designs: **a** double-ended, annular gap restriction; **b** double-ended, cylindrical hole restriction; **c** single-ended, annular gap restriction; **d** double-ended, cylindrical hole restriction) used as a starting point to design; **e** STF-based adjuster

because of the cavitation explained in the next paragraph, while the limiter's role is to avoid too large elongation. Due to the upper air spring, the upper chamber is pressurized continuously, which causes elongation of the adjuster. From the engineering perspective, applying a membrane instead of a piston can decrease the leakage, while replacing the air spring with a helical spring enhances reliability.

Let us imagine that downward force is applied on the piston rod while the cylinder stays at rest. Then the pressure increases in the lower chamber and decreases in the upper chamber. If the force builds up abruptly, the pressure of the upper chamber can go below environmental pressure, and cavitation can happen, especially in the case of low velocities, see Dixon (2008), Golinelli and Spaggiari (2017), Castellani et al. (2017), Gao et al. (2020). In damper applications, a helical spring is often installed parallel with the damper (see automotive examples Dixon 2008), the lower chamber also has underpressure in rest. Cavitation can be critical by employing STF because the structure (and hence the rheology) of the fluid can be changed significantly, which causes modifications in the adjuster's operation. Therefore in our design, the compensating secondary piston is connected to the other chamber. Moreover, the whole device is continuously pushed; thus, not only the upper chamber is continuously pressurized, but also the lower chamber. These considerations ensure a cavitation-free operation.

2.2 The proposed adjuster

Our adjuster is to be built between an actuator (usually a pneumatic cylinder, e.g., brake cylinder Nunney (2007), clutch actuator Nunney (2007)) including a helical spring to maintain connection between the elements and a stiff, but elastic element (e.g., brake disk Nunney (2007), clutch spring Nunney (2007)). For obtaining more realistic results, the adjuster is also modeled in this study, see Fig. 4. The single-acting pneumatic cylinder is charged with compressed air with a pressure of 8 bar and temperature of 20 °C or vented to the environment. The proper control of the load and exhaust valves' opening and closing provides the piston's desired position. The adjuster would be mounted between the piston of the cylinder and the counter spring.

A typical process can be seen in Fig. 2. In the equilibrium state, the spring forces are balanced (Fig. 2a). If the desired position is changed to x_d value, then load valves open suddenly, and the desired position is reached (Fig. 2b). While holding the fixed desired position, the adjuster starts to compress (denoted by x_c) due to the high compression; hence the actuator position should be increased further (Fig. 2c). Due to the wear, the null position of the counter spring alters with w ; hence, the actuation displacement increases (Fig. 2d). Finally, the desired position retracts to the initial state, the exhaust

Table 1 Operating environments of the self-adjuster

Name	Notation	Value
Actuator spring stiffness	s_{cyl}	1.02 N/mm
Pre-tension force of the actuator	$s_{cyl} \Delta l_{0,cyl}$	150 N
Supply pressure	$\max(p_{cyl})$	8 bar
Actuator piston area	A_{cyl}	7850 mm ²
Counter spring stiffness	s_1	300 N/mm

Fig. 2 A typical process of an adjuster. The process starts from equilibrium (a). During actuation, the cylinder's piston moves with x_d desired position, and hence the counter spring is compressed (b). Due to high forces, the adjuster compresses with x_c , and the piston moves with that displacement further (c). Then the friction disks become worn, and the counter spring's null position alters with w (d). Finally, the actuation is over, and the piston tends to the initial state (e). However, the equilibrium position is displaced with that $x_c + w$ length. The role of the adjuster is to compensate that length f)

valves of the cylinder open, and the pressure decreases to the environmental pressure. Since the adjuster compresses and the null position of counter spring is displaced, the piston position will differ from the initial one (Fig. 2e). If the adjuster works correctly, the upper chamber's pressure is higher than the downstream chamber's; thus, the fluid starts to flow from the upper chamber to the lower one. Thus, the adjuster elongates, and the actuator piston reaches its initial state (Fig. 2f).

2.3 Design requirements and objectives

As mentioned earlier, the adjuster has two main functions: (1) be sufficiently stiff during actuation and (2) compensate the dead point modification, which is the sum of the compression during actuation and the wear. If the gap is too small or the fluid is too dense, the adjuster will be very stiff during actuation but not flexible enough during the adjusting period. On the other hand, if the gap is too large or the fluid is too “soft”, the adjuster will be too flexible during actuation. Hence, a compromise must be reached between these two objectives.

We now provide an application through which we will present the design process. The functional requirements of the particular application, e.g., data of the operating environment, including load and the desired adjusted length, can be found in Table 1.

The most critical requirements for the self-adjuster concern the speed of the adjustment during the unloaded state and the unwanted position creep during actuation, i.e., the loaded states. These requirements appear both in the form of constraints as well as objectives. As a constraint, the most crucial requirement for the self-adjuster is that it shall elongate when it is unloaded at least the same amount as it was shortened during the actuation period. For this, a total stroke of actuation shall be considered from the point when only the pre-tension force is applied until reaching the full load. After that begins the compensation phase, which lasts for maximum 20 s, and only the pre-tension force is applied. This test period is depicted in Fig. 5.

Besides formulating the self-adjustment capabilities as hard limiting constraints, they are also subjected to optimization. The two objectives are maximizing the self-adjustment speed in the unloaded state and minimizing the unwanted shortening during loaded states. The two objective functions and constraints (Table 4) surmise a Pareto front as depicted in Fig. 15.

In the first case, the simulation started from the initial state, and the desired position changed abruptly with 20mm. In the second case, the initial condition was the same, the desired position is kept constant, and the wear increased abruptly. The output functions of the optimization are the compression during actuation and the elongation during adjustment. Based on the obtained Pareto front, the final design could be determined.

3 Methods

The governing equations comprise a six-degree-of-freedom system of ordinary differential equations (ODEs). The effect of the non-Newtonian fluid is taken into account with the $\Delta p(Q)$ function, where Δp is the pressure difference between

the chambers and Q is the volume flow rate through the chambers. These $\Delta p(Q)$ functions have analytical forms and derived for well-known non-Newtonian fluids, e.g., Zhang et al. (2008) for Bingham, Chhabra and Richardson (2011) for power-law and Kemerli and Engin (2021) for Hershel–Bulkley model.

However, only the authors’ previous researches (Nagy-György and Hős 2021, 2019) give a generally applicable technique to determine $\Delta p(Q)$ function using directly rheological measurement data without any model fitting. The effect of the shear thickening rheology is derived from the simplified Navier-Stokes equation, and the obtained $\Delta p(Q)$ functions are substituted into the ODEs for 6 different rheologies. This derivation is checked through CFD simulations. Then, multi-objective optimization is performed with a genetic algorithm using the built-in *gamultiobj* Matlab function on the previously mentioned ODES.

Although the coupled fluid-and-solid dynamic method (substituting flow equations into equations of the solid body motion) is validated experimentally (Zhang et al. 2008; Kemerli and Engin 2021) for non-Newtonian fluid-filled damper, we check its accuracy utilizing time-dependent CFD simulations.

The optimisation procedure requires different numerical tools, which will be presented in the following subsections. To strength the cohesion between the subsections, the whole optimisation procedure is illustrated with a flow chart in Fig. 3.

3.1 Governing equations

According to the test specification in Sects. 2.2 and 2.3, the behavior of the self-adjuster during the virtual test can be characterized by solving equations of motion corresponding to the different bodies noted with (1), (2) and (3) in Fig. 4. The forces acting on the bodies can be illustrated by the free body diagrams, shown in Fig. 4. Newton’s second law for the first body is

$$\sum F_1 = F_{pu1} + F_{pd1} - F_{s1} = m_1(\ddot{x}_1 + \ddot{x}_2), \quad (1)$$

where $F_{pu1} = p_u A_{u1}$ is the upstream pressure p_u force acting on the surface A_{u1} , $F_{pd1} = p_d A_{d1}$ is the downstream pressure p_d force acting on the surface A_{d1} and F_{s1} is the counter spring force. On the right hand side m_1 denotes the mass of the adjuster and actuator piston. $\ddot{x}_1$ is the second time derivative of x_1 dimension, which is the acceleration of the adjuster compression or elongation, while $\ddot{x}_2$ is the second time derivative of x_2 dimension, which is the acceleration of the actuator piston. Important to note that, $\ddot{x}_1$ is the *relative* acceleration of body (1) relative to body (2), thus the absolute acceleration of body (1) is $(\ddot{x}_1 + \ddot{x}_2)$.

For the second body we have

Fig. 3 Flow chart of the optimisation procedure

$$\sum F_2 = F_{p,cyl} + F_{s,cyl} + F_{pd2} - F_{s3} - F_{pu2} = m_2 \ddot{x}_2. \quad (2)$$

The *cyl* subscript refers to the data of the actuator (which is a pneumatic cylinder), in order not to be confused with actuating process. The actuator force divided into $F_{p,cyl} = p_{cyl} A_{cyl}$ actuator pressure p_{cyl} force acting on the actuator piston

Fig. 4 Free-body diagram of the parts

surface A_{cyl} and to the $F_{s,cyl} = s_{cyl}(\Delta l_{0,cyl} - x_2)$ actuator spring force, where s_{cyl} is the actuator spring stiffness and $\Delta l_{0,cyl}$ is the initial compression. The fluid forces acting on body (2) are the $F_{pd2} = p_d A_{d2}$ downstream pressure p_d force acting on the surface A_{d2} and $F_{pu2} = p_u A_{u2}$ upstream pressure p_u force acting on the surface A_{u2} . The force of the spring (3) is $F_{s3} = s_3(\Delta l_{0,3} + x_3)$, where s_3 is the spring stiffness and $\Delta l_{0,3}$ is the initial compression of spring (3). These notations are represented in Fig. 4.

Finally, Newton's second law for the third body has the form of

$$\sum F_3 = F_{s3} - F_{pd3} = m_3(\ddot{x}_2 + \ddot{x}_3), \quad (3)$$

where F_{s3} is the above-defined spring force and $F_{pd3} = p_d A_{d3}$ is the pressure p_d force acting on piston surface A_{d3} . $\ddot{x}_3$ is the second time derivative of dimension x_3 , which is the relative acceleration (3) piston corresponding to body (2).

The adjuster has 5 geometrical parameter (d_i, d_o, D_d, D_u and L), the rest of the parameters are

$$\begin{aligned} A_{d1} &= \frac{d_i^2 \pi}{4}, & A_{u1} &= \frac{D_u^2 \pi}{4} - A_{d1}, & A_{d2} &= \frac{D_d^2 \pi}{4} - A_{d1}, \\ A_{u2} &= \frac{D_u^2 \pi}{4} - A_{d1}, & A_{d3} &= \frac{D_d^2 \pi}{4}. \end{aligned} \quad (4)$$

The wall thickness is estimated to be uniformly 10mm, with which the volumes and masses can be easily calculated.

The shear thickening fluid is supposed to be incompressible; thus, the volume change in the upstream chamber equals the volume change in the downstream chamber. This assumption gives

$$A_{u1} \Delta x_1 = A_{d3} \Delta x_3 \quad (5)$$

$$A_{u1} \dot{x}_1 = A_{d3} \dot{x}_3 = Q_{gap} \text{ and} \quad (6)$$

Fig. 5 Typical solutions of the differential equation system of (13) and (14). For actuating case linear ramp cylinder pressure ($p_{\text{cyl}}(t)$) and constant wear $w = 39.5$ mm (a), for adjusting constant cylinder pres-

sure and linear ramp wear from $w = 39.5$ mm to $w = 40$ mm were set (b). For the real application case, previous cases are mixed (c)

Fig. 6 Rheological measurement data of the investigated liquids (Soutrenon and Michaud 2014; Tian and Nakano 2017; Sun et al. 2013; He et al. 2015; Miao et al. 2018; Serrano-Aguilera et al. 2016) corresponds to different colors. The discrete measurement data represented by the symbols were used directly as transport properties in CFD simulations. For further material properties, see Table 2. The linestyles corresponds to the different regions of rheological curve: Region 1 - dashed, Region 2 - continuous, Region 3 - dotted

$$A_{u1}\ddot{x}_1 = A_{d3}\ddot{x}_3. \quad (7)$$

Moreover, $A_{u1}\dot{x}_1 = A_{d3}\dot{x}_3 = Q_{\text{gap}}$ is the volume flow rate through the gap. The pressure difference $p_u - p_d = \Delta p(Q_{\text{gap}})$ is the function of that volume flow rate, i.e.

$$p_u - p_d = \Delta p(A_{u1}\dot{x}_1), \quad (8)$$

which can be calculated with the authors' previously published formula for Poiseuille flow in Nagy-György and Hős (2019), Nagy-György and Hős (2021). The main steps are summarized in Sect. 3.2, for a more detailed description, we refer to Nagy-György and Hős (2021).

Let us define the surface ratio as $\mathcal{A}_1 := \frac{A_{u1}}{A_{d3}}$ surface ratio. From (3) p_d and p_u can be expressed using (7) and (8)

$$p_d = \frac{1}{A_{d3}} [F_{s3} - m_3(\ddot{x}_2 + \mathcal{A}_1\ddot{x}_1)] \quad (9)$$

and

$$p_u = p_d + \Delta p(\mathcal{A}_1 \ddot{x}_1) = \frac{1}{A_{d3}} [F_{s3} - m_3(\ddot{x}_2 + \mathcal{A}_1 \ddot{x}_1)] + \Delta p(\ddot{x}_1 A_{u1}) \quad (10)$$

$x_3(t)$, $p_u(t)$, and $p_d(t)$ are expressed as a function of $x_1(t)$ and $x_2(t)$ using constraints (5), (6), (7), and (8). Then, (10) and (9) can be substituted into Eqs. (1) and (2) yielding a second order linear equation system for $\ddot{x}_1$ and $\ddot{x}_2$:

$$\begin{bmatrix} m_1 + \mathcal{A}_1 \mathcal{A}_2 m_3 & m_1 + \mathcal{A}_2 m_3 \\ \mathcal{A}_1 \mathcal{A}_3 m_3 & m_2 + \mathcal{A}_3 m_3 \end{bmatrix} \begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{bmatrix} = \begin{bmatrix} -F_{s1} + \mathcal{A}_2 F_{s3} - A_{u1} \Delta p \\ F_{cyl} + (\mathcal{A}_3 - 1) F_{s3} - A_{u2} \Delta p \end{bmatrix}, \quad (11)$$

where

$$\mathcal{A}_2 = \frac{A_{u1} - A_{d1}}{A_{d3}}, \quad \mathcal{A}_3 = \frac{A_{d2} - A_{u2}}{A_{d3}}. \quad (12)$$

The acceleration $\ddot{x}_1$ and $\ddot{x}_2$ can be expressed by solving (11). We have

$$\begin{aligned} \ddot{x}_1 &= \frac{1}{den} [a_1 F_{cyl} + a_2 F_{s1} + a_3 F_{s3} + a_4 \Delta p(\ddot{x}_1 A_{u1})] \\ &= \frac{1}{den} [a_1 (p_{cyl} A_{cyl} + s_{cyl}(l_{0,cyl} - x_2)) + a_2 s_1(x_1 + x_2 - w) \\ &\quad + a_3 s_3(l_{0,3} - \mathcal{A}_1 x_1) + a_4 \Delta p(\ddot{x}_1 A_{u1})] \end{aligned} \quad (13)$$

and

$$\begin{aligned} \ddot{x}_2 &= \frac{1}{den} [b_1 F_{cyl} + b_2 F_{s1} + b_3 F_{s3} + b_4 \Delta p(\ddot{x}_1 A_{u1})] \\ &= \frac{1}{den} [b_1 (p_{cyl} A_{cyl} + s_{cyl}(l_{0,cyl} - x_2)) + b_2 s_1(x_1 + x_2 - w) \\ &\quad + b_3 s_3(l_{0,3} - \mathcal{A}_1 x_1) + b_4 \Delta p(\ddot{x}_1 A_{u1})], \end{aligned} \quad (14)$$

where the denominator is defined as

$$den = m_1 m_2 + m_1 m_3 (1 - \mathcal{A}_1) \mathcal{A}_3 + m_2 m_3 \mathcal{A}_1 \mathcal{A}_2 \quad (15)$$

and the coefficients are

$$\begin{aligned} a_1 &= -m_1 - m_3 \mathcal{A}_2 \\ a_2 &= -m_2 - m_3 \mathcal{A}_3 \\ a_3 &= m_1 - m_1 \mathcal{A}_3 + (m_2 + m_3) \mathcal{A}_2 \\ a_4 &= A_{u2}(m_1 + m_3 \mathcal{A}_2) + A_{u1}(m_2 + m_3 \mathcal{A}_3) \end{aligned} \quad (16)$$

and we also have

$$\begin{aligned} b_1 &= m_1 + m_3 \mathcal{A}_1 \mathcal{A}_2 \\ b_2 &= m_3 \mathcal{A}_1 \mathcal{A}_3 \\ b_3 &= (-1 + \mathcal{A}_3) m_1 - m_3 \mathcal{A}_1 \mathcal{A}_3 \\ b_4 &= -A_{u2} m_1 + \mathcal{A}_1 (\mathcal{A}_3 A_{u1} + \mathcal{A}_2 A_{u2}) m_3 \end{aligned} \quad (17)$$

Equations (13) and (14) are solved by *ode23s* built-in *Matlab* stiff ordinary differential equation solver with $x_1(t=0) = 30$ mm and $x_2(t=0) = 10$ mm equilibrium initial conditions.

The adjuster should have two different functions (stiff during actuation, flexible during released state), which are investigated in two different optimization cases. In the first case the pressure in the cylinder p_{cyl} is increased from 0 to 8 bar with a linear ramp function (see Fig. 5a) while the $w(t) = 39.5$ mm wear is kept constant. The objective function is the $o_1 := \Delta x_1$ compression of the adjuster. In the second case cylinder pressure $p_{cyl} = 0$ bar is kept constant and the wear $w(t)$ is increased from 39.5 to 40 mm with a linear ramp function as 5 shows. In that case, the adjuster should elongate, and the prescribed $\Delta w = 0.5$ mm wear should be compensated. We define the adjusting time 25 s, so the second objective is the length, which can not be compensated in the 20 s, i.e., $o_2 := w - x_1(t=25)$. Typical solutions and further explanation of objectives are in Fig. 5c and e. After the optimization, the obtained optimal designs are checked with a real application-like case. In that case, the previously defined functions are represented in Fig. 5.

3.2 Flow equations

Non-Newtonian flow through an annular gap is not only a fundamental problem of analytical fluid mechanics, but it is also relevant in numerous practical applications, for example damper modeling (Lin et al. 2018; Zhou et al. 2016; Zhang et al. 2008; Hong et al. 2008; Golinelli and Spaggiari 2017; Kemerli and Engin 2021) or drilling (Krishna et al. 2020). In Nagy-György and Hős (2021), the authors of the present study proposed a technique to cope with arbitrary rheology to determine the $\Delta p(Q)$ characteristics of the flow through a gap or a circular hole.

The method aims to analyze the flow in arbitrary rheology with shear stress vs shear rate curve denoted by $\tau = g(\dot{\gamma})$. For example, in the case of Newtonian fluid we have $\tau = g(\dot{\gamma}) = \mu \dot{\gamma}$. Let us introduce the $f(\tau)$ shear rate-shear stress function which is the inverse of the above-defined $g(\dot{\gamma})$ function. e.g., $f(\tau) := g^{-1}(\dot{\gamma})$ gives $\dot{\gamma} = f(\tau) = \frac{\tau}{\mu}$ for Newtonian fluid.

Besides the rheology we need the geometrical data of the adjuster to calculate $\Delta p(Q)$ characteristics. The gap dimensions are the d_i , d_o diameters and the L length (see Fig. 4), from that the W circumference and the h half width of the gap are

$$W = \frac{(d_o + d_i)\pi}{2}, \quad \text{and} \quad (18)$$

$$h = \frac{d_o - d_i}{2}. \quad (19)$$

The basic idea of the derivation is to integrate the simplified momentum equation with respect to τ local shear stress instead of the commonly used y spanwise coordinate, using the fact that the local shear stress is proportional to the y coordinate. When the first and the second integrate of $f(\tau)$ function are denoted by

$$F(\tau) = \int_0^\tau f(\tilde{\tau}) d\tilde{\tau}, \quad (20)$$

$$S(\tau) = \int_0^\tau F(\tilde{\tau}) d\tilde{\tau} \quad (21)$$

then the volume flow rate Q through the gap is

$$Q = \frac{Wh^2}{\tau_p^2} [2\tau_p F(\tau_p) - 2S(\tau_p)], \quad (22)$$

where

$$\tau_p = \frac{\Delta p h}{L} \quad (23)$$

is the shear stress at the walls.

Equation (22) allows the computation of the volume flow rate for an arbitrary generalized Newtonian rheology. The first step is to integrate $f(\tau)$ function twice either analytically or, possibly, numerically using the trapezoidal method with spreadsheet software to obtain $F(\tau)$ function after the first, and $S(\tau)$ function after the second integration. Then, the $F(\tau_p)$ and $S(\tau_p)$ values can be read at $\tau = \frac{\Delta p}{L}h$ for every corresponding pressure difference values, and the volume flow rate value can be obtained in those points substitution into (22). Based on this method, the $Q(\Delta p)$ plot can be drawn, and for a given $Q = A_{ul}\dot{x}_1$ volume flow rate value the $\Delta p = p_u - p_d$ value can be obtained by interpolation.

Let rearrange Eq. (22) to the form

$$\tilde{Q} := \frac{Q}{Wh^2} = \frac{1}{\tau_p^2} [2\tau_p F_{odd}(\tau_p) - 2S_{odd}(\tau_p)]. \quad (24)$$

The above-defined $\tilde{Q} := \frac{Q}{Wh^2}$ quantity in dimension $1/s$ depends only on the fluid rheology. For a given rheology the $\tilde{Q}(\tau_p)$ function can be determined directly using Eq. (24) before starting optimisation and even before solving any ODES. In that case, the $\tilde{Q}(\tau_p)$ curve is input data of the (13),(14) ODES, and for a given $\tilde{Q} = \frac{A_{ul}}{Wh^2}\dot{x}_1$ the $\tau = \frac{\Delta p}{L}h$ value is read. We considered six different rheologies represented by the upper panel of Fig. 6. The corresponding relative flow characteristics were checked by means of CFD simulations. The comparison can be seen in Fig. 7.

3.3 CFD model

Up to the authors' best knowledge, there is no measurement data on STF-based self-adjuster. Therefore, the accuracy of the analytical model is proved via CFD simulations. Formula (24) is validated against steady-state simulations, while transient simulations were also performed to check the behavior of the final optimized design.

The corresponding CFD simulations were performed using OpenFOAM Version 6. Two-dimensional flow was assumed to reduce the computational cost, and only a 5° circular slice was modeled. The geometry's dimensions can be seen in Fig. 8. The mesh was generated by the built-in *blockMesh* function, which divides the whole domain into hexahedron cells. To eliminate the discretization error, mesh independence analysis was performed with four different mesh sizes: 23.7k, 56.2k, 133.4k, 316.2k elements.

The analysis performed using setting, with which the highest velocity values are obtained, hence material properties of STF5 are used and $d_i = 5.24$ mm, $L = 10$ mm, $d_o = 6$ mm, $D_d = 24$ mm, $D_u = 30$ mm dimensions applied with the inlet velocity 1 mm/s. The upper panel of Fig. 10 shows the obtained values corresponding to the different mesh sizes, while the lower panel of Fig. 10 represents the relative error comparing to the chosen mesh.

The relative difference in the obtained force values was less than 1% in the case of the three finest meshes. Thus, the mesh with 133.4k cells was chosen for the following simulations, shown in Fig. 9. The inlet boundary condition of the steady-state simulations was a constant inlet flow velocity in the range of 0.01–1 mm/s, while 0 Pa relative pressure is set at the outlet. At all other boundaries, a no-slip wall condition was employed.

Fig. 7 Obtained relative flow characteristics using Eq. (24) and CFD model described in Sect. 3.3. The CFD simulations were performed with two different geometry. In the first case (*CFD1*) the inner diameter was $d_i = 5.24$ mm, while $d_i = 5.49$ mm was in the second case (*CFD2*). The other dimensions were the same in both cases: $L = 10$ mm, $d_o = 6$ mm, $D_d = 24$ mm and $D_u = 30$ mm (see Fig. 4). The results of the CFD simulations lie on the analytical curve in all cases

Fig. 8 Sketch of the simplified damper geometry with the corresponding boundary conditions. Two types of boundary conditions were used for the inlet: a given constant velocity for the steady case and a $p(t)$ pressure function in time for the transient simulations. The $p(t)$ function obtained from the analytical results. The dimensions (d_i, d_o, D_u, D_d, L) for the steady case are represented in the brackets, while the values for transient case are plotted in Fig. 14

During the simulations, the *strainRateFunction* transport properties function was used, which is able to calculate the actual value of viscosity from the measurement data directly via interpolation. For the steady-state simulations, the shear rate-shear stress points of STF1–STF6 were used, depicted by Fig. 6. The authors wish to emphasize here that a fundamental prerequisite for both the analytical and CFD model is that neither of them requires analytical rheology model fit but use “raw” (point-wise) $\tau(\dot{\gamma})$ data.

Fig. 9 Mesh of the investigated damper with an annular gap restriction. The mesh contained only one cell layer in the tangential dimension using the 2D simplification. The mesh contained only hexagonal elements and is refined near the gap

Csizmadia and Hős (2013), Csizmadia and Hős (2014), Bíbok et al. (2020), Csizmadia and Till (2018) show that the results obtained by the SST model have a reasonable agreement with the measurement data for non-Newtonian pipe flows; therefore, $k-\omega$ SST model was used with the recommended settings of Greenshields (2017).

For solving continuity and momentum equations, the smooth Gauss-Seidel method was used. Linear Gauss and limited, bounded Gauss schemes were applied as gradient and divergence scheme, while Gauss linear orthogonal scheme was chosen as the Laplacian scheme. The interpolation scheme was linear (Greenshields 2017). The convergence was found to be highly sensitive to solvers' tolerance, especially near the thickening effect, where the pressure drop increases abruptly. For the thinning region $1e-10$ tolerance was acceptable, however, in the thickening regime only $1e-15$ tolerance and relaxation factors of 0.3 provided converged solution. For the transient simulations $1e-15$ tolerance and $CFL < 0.5$ condition were used.

3.4 Materials

A wide range of papers are available in the literature concerning STF manufacturing and measuring its rheology, out of which we have chosen six different liquids, consisting of different types of base liquids and the different particles. The shear thickening rheologies can be characterized by different properties, e.g., (a) $\mu(\dot{\gamma} = 1)$ zero viscosity, (b) critical shear rate, where the shear thickening effect starts, and (c) shear thickening ability (the ratio of the minimum and maximum viscosity). To illustrate the effect of these shear thickening properties on the adjuster's behavior, we aim to choose different liquids with different shear thickening properties. Moreover, a further aspect was to use easy-to-manufacture and cost-efficient liquids to support industrial application

Fig. 10 Results of the mesh independence analysis. The upper panel shows the obtained force values, while the lower panel represents the relative difference comparing to the chosen mesh. The mesh independence analysis was performed in the simulation steady-state simulation point, where the highest velocity values are obtained, hence using STF5 and $d_i = 5.24$ mm, $L = 10$ mm, $d_o = 6$ mm, $D_d = 24$ mm and $D_u = 30$ mm dimensions

later. Therefore, most of the particles and the selected fluids are commercially available products.

The particle sizes are between 12 and 500 nm, so between nano and micro scales, while the particle distribution is also varied. *STF1*, *STF2* and *STF3* consist monodispersed distribution, while *STF4*, *STF5* and *STF6* have commercially available products of *Novtech* or *Sigma-Aldrich* company. Their distribution is close to normal distribution. For more detail, see Aerosil and Sigma-Aldrich catalog. The base liquid is also varied according to the molar mass from 60 to 400 g/mol, hence ethylene-glycol (EG), polyethylene-glycol with a molar mass of 200 g/mol (PEG200) and 400 g/mol (PEG400), and polypropylene-glycol with a molar mass of 400 g/mol is used.

3.5 Genetic algorithm

Genetic algorithms are heuristic optimization approaches that are able to find the close-to-global optimal solution. Mitchell (1998), Eiben and Smith (2003) They are widely used for shape optimization in engineering problems, mostly coupled with any simulation computations. (Sun et al. 2007; Woon et al. 2001; Tian and Li 2020; Kajtaz 2019; Yıldız and Lekesiz 2017; Yoo et al. 2019) The benefit of genetic algorithms is that they are robust and work on almost any kind of problems; however, the convergence in the direction of the global optimum (optima) cannot be guaranteed, thus the correct (and often problem-specific) parameter settings play a central role in convergence behavior. (Mitchell 1998; Gibbs et al. 2010; Montero et al. 2014)

In the present study, the optimization aims to find the best geometrical design. Hence, the target is to find a solution that meets the requirements regarding functionality and cost rather than finding a global optimum, which makes the genetic algorithm a reasonable choice for optimizing the self-adjuster.

There are two competing optimization aims: (1) the optimal design shall elongate “automatically” in the unloaded state to exhibit the self-adjustment feature, and (2) it shall be rigid enough during the loaded state in order to transfer forces during actuation. These requirements

result naturally in a multi-objective optimization problem, where the two objectives are the self-adjustment time and the unwanted shortening, both to be minimized.

For multi-objective optimization problems, the objective functions can be aggregated (Deb 2001; Wang et al. 2020; Marler and Arora 2004; Osyczka and Kundu 1995) or a real multi-objective optimization can be performed, which results in a Pareto front of non-dominated solutions rather than one single solution (Deb 2001; Marler and Arora 2004; António 2013; Jafaryeganeh et al. 2020). A non-dominated solution means that there is no other solution that is better in all of the objective functions. After the optimization, the winning design can be selected from the set of these solutions (from the Pareto front) by a higher decision (typically performed by a human).

Since the numerical apparatus for the present research was built up in Matlab environment, the *gamultiobj* multi-objective genetic algorithm optimization solver was used Natick (2017). This is based on the *NSGA-II* technique (Non-Sorting Genetic Algorithm-II) Deb et al. (2002) and proven to be capable of solving real-life design and shape optimization problems in various field of engineering. (Dehghani et al. 2020; Shojaeefard et al. 2019; Zandavi and Pourtakdoust 2018)

Our optimization problem can be described as follows. The free variables to be optimized are given in Table 3. The two objective functions to be minimized are

$$o_1(d_i, d_o, D_u, D_d, L) = x_1(t = 0 \text{ s}) - x_1(t = 5 \text{ s})$$

with

$$p_{\text{cyl}}(t) = \begin{cases} 0 \text{ bar} & \text{if } t \leq 1. \\ 7.64 \text{ bar} \cdot t/(0.2 \text{ s}) & \text{if } 1 < t < 1.2, \\ 7.64 \text{ bar} & \text{if } 1.2 \leq t, \end{cases} \quad (25)$$

$$w(t) = 39.5 \text{ mm}$$

and

$$o_2(d_i, d_o, D_u, D_d, L) = 30.5 \text{ mm} - x_1(t = 25 \text{ s})$$

with

Table 2 Material properties of the fluids

STF	Src.	Base liquid	Average diameter, nm	v/v%	Manufacturer
1	Sun et al. (2013)	EG	500	36	Authors
2	He et al. (2015)	EG	60	43	Authors
3	Soutrenon and Michaud (2014)	PEG200	500	68	Authors
4	Miao et al. (2018)	PEG400	12	30	Sigma-Aldrich
5	Tian and Nakano (2017)	PPG400	40	45	Aerosil OX50
6	Serrano-Aguilera et al. (2016)	PPG400	12	10	Aerosil R816

Table 3 Bounds of the design variable for multi-objective optimisation

Nr.	Design variable name	Notation	Lower bound (mm)	Upper bound (mm)
1	Gap inner diameter	d_i	1	100
2	Gap outer diameter	d_o	2	100
3	Lower diameter	D_d	5	200
4	Upper diameter	D_u	5	200
5	Gap Length	L	1	200

$$p_{cyl}(t) = 0 \text{ bar}$$

$$w(t) = \begin{cases} 39.5 \text{ mm} & \text{if } t \leq 1 \\ 39.5 \text{ mm} + 0.5 \text{ mm} \cdot t/(2 \text{ s}) & \text{if } 1 < t < 2 \\ 40 \text{ mm} & \text{if } 2 \leq t, \end{cases} \quad (26)$$

where $x_1(t)$ is the solution of Eqs. (13)–(17) with the corresponding $p(t)$ and $w(t)$ functions. The typical x_1 solutions and the definition of o_1 and o_2 is illustrated in Fig. 5d and e, while the corresponding $p(t)$ and $w(t)$ functions are shown in 5 a) and b). In (25) and (26) d_i , d_o , D_u , D_d and L represents the design variables, which are the gap inner diameter, the gap outer diameter, the lower chamber's diameter, the upper chamber's diameter and the gap length.

The search space had to be chosen very carefully since highly skewed designs result in numeric computational problems, see Sect. 3.6.

3.6 Definition of design variable range

This section provides a set of possible constraints and demonstrates how to find the possible set of solutions (Pareto front) and choose the final design by employing higher-level aspects. Obviously, our choice of design variables is problem-dependent.

3.6.1 Linear constrains

The assembly space limits the adjuster's size, and hence it provides the bounds of the design variables. These bound are summarized in Table 3. Moreover, there are feasibility constraints between the diameters:

1. The gap's outer diameter should be larger than the inner one's with a realistic tolerance: $d_o \geq d_i + 0.001 \text{ mm}$.
2. The gap's outer diameter should be smaller than the downstream chamber diameter: $D_d \geq d_o + 5 \text{ mm}$.
3. The upstream chamber diameter should be smaller than the downstream chamber diameter with a realistic wall thickness $s = 5 \text{ mm}$: $D_u \geq D_d + 5 \text{ mm}$.

These linear constrains can be described in the mathematical form of

$$\begin{bmatrix} 1 & -1 & 0 & 0 & 0 \\ 0 & 1 & -1 & 0 & 0 \\ 0 & 0 & 1 & -1 & 0 \end{bmatrix} \begin{bmatrix} d_i \\ d_o \\ D_d \\ D_u \\ L \end{bmatrix} < \begin{bmatrix} -0.001 \text{ mm} \\ -5 \text{ mm} \\ -5 \text{ mm} \end{bmatrix}. \quad (27)$$

3.6.2 Non-linear constrains

The fluid mechanical modeling approach presented in Sect. 3.2. is reasonable accurate only in the case of small relative gap sizes h/L and h/d_o . According to the previous studies Filip and David (2013) these ratios are set arbitrarily to $h/L < 0.05$ and $h/d_o < 0.1$.

On the other hand, the numerical scheme of the applied ODE solver limits the design variable range. The condition number of the governing differential equation system and the suitable numerical solver are strongly affected by the flow model. In the case of low pressure losses (large gap and/or low viscosity fluid), the adjuster compresses abruptly, which requires traditional solvers, e.g., the 4–5th Runge-Kutta-Fehlberg method. However, high pressure losses (small gap and/or high viscosity fluid) the adjuster's compression is very small (in the range of nanometers, three orders of magnitude lower than the 20 mm actuation length), which requires the use of stiff solvers. Moreover, Eqs. (13) and (14) need to be solved not only a few times but thousands of times during the optimization with different condition numbers.

To increase the reliability and accuracy of the multi-objective optimization, the proper definition of the design variable domain is essential. Two criteria are used to define the design variable domain: (1) the compression velocity of the adjuster should be lower than $v_{\text{crit,act}} = 0.01 \text{ mm/s}$ during a stationary load of actuation force ($F_{\text{act}} = 6000 \text{ N}$), and (2) the adjusting velocity of the adjuster should be higher than $v_{\text{crit,adj}} = 0.0001 \text{ mm/s}$ due to the equilibrium force change ($F_{\text{adj}} = 10 \text{ N}$ caused by the wear).

Using both of the criteria, firstly the v_{act} , v_{adj} velocity values need to be read from $F(v)$ graph corresponding to the actuation ($F_{\text{act}} = 6000 \text{ N}$), and adjusting ($F_{\text{adj}} = 10 \text{ N}$) force. In Fig. 11a–h curves corresponds to different design variable combinations, on which the circles represent the $(v_{\text{act}}, F_{\text{act}})$ and squares the $(v_{\text{adj}}, F_{\text{adj}})$ points. Curves g) and h) does not satisfy $v_{\text{adj}} > v_{\text{crit,adj}}$ and $v_{\text{act}} < v_{\text{crit,act}}$ conditions, thus those design variable combinations are not accepted.

The $F(v)$ curve is divided into three regions. Region 1 is the first region, where the shear thinning region of the rheological curve works (dashed in Fig. 11). Region

2 is the middle region, where the fluid thickens and the flow curve increase abruptly (continuous line in Fig. 11). Region 3 is third region, where the fluid already thickened (dotted in Fig. 11).

Depending on the d_i, d_o, D_d, D_u, L design variables, cross points of $F(v)$ curve with $F_{crit,1}$ horizontal line (point $P_s @ (v_{act}, F_{act})$, squares in Fig. 11) and $F_{crit,2}$ horizontal line (point $P_c @ (v_{act}, F_{act})$, circles in Fig. 11) can be get to the different regions on the $F(v)$ curve:

- both P_s and P_c in Region 3 (dotted)
- P_s in Region 3 and P_c in Region 2 (cont.)
- P_s in Region 3 and P_c in Region 1 (dashed)
- both P_s and P_c in Region 2 (cont.)
- P_s in Region 2 and P_c in Region 1 (dashed)
- both P_s and P_c in Region 1 (dashed)

The shear thickening fluid-filled self-adjuster works less effective in the (a) (a on Fig. 11) and (f) (f on Fig. 11) cases, while the most effective way in the (c) case, hence only the (b–e) (b–e on Fig. 11) cases will be considered later.

The non-linear constraints are summarized by Table 4 and the following equations:

$$\begin{aligned} \frac{d_o - d_i}{4L} &< 0.01 \\ \frac{d_o - d_i}{4d_o} &< 0.1 \\ v(6000 \text{ N}) &< 0.1 \text{ mm/s} \\ v(10 \text{ N}) &> 0.001 \text{ mm/s} \\ v(6000 \text{ N}) &\notin \text{Region 1 in Fig. 11} \\ v(10 \text{ N}) &\notin \text{Region 3 in Fig. 11.} \end{aligned} \quad (28)$$

Summarizing the complete task, we have a multi-objective optimisation problem for objective functions o_1 and o_2 (25, 26) with respect to the constraint system described in (27, 28).

Fig. 11 Representation of the non-linear constraints of the optimization. The lines represent the $F(v)$ curve corresponding to different (d_i, d_o, D_d, D_u, L) design variable combinations. The adjuster is too rigid during adjusting in case g) and too flexible during actuation in case a). f represents the case when the adjuster works in Region 1 (dashed) and a) when the adjuster works in Region 3 (dotted). The acceptable cases for the multi-objective optimization are b–e

4 Results

4.1 Optimal fluid

The optimization results corresponding to different rheologies are represented by Pareto fronts in Fig. 12. In the ideal case, all points are in origin, since at that point, both functions are satisfied. The closer this point to the origin is, the better the design meets the requirements. According to the Fig. 12 the Pareto front of *STF3* and *STF5* are the closest, while *STF4* is the farthest from the origin.

Each Pareto point corresponding to different design. To get more precise picture about the designs, mixed simulations (see third column of Fig. 5) are performed to investigate the adjusting behavior of each design. The flexibility characterized with the adjuster length change $dx_{full} = x_1(t = 25 \text{ s}) - x_1(t = 0 \text{ s})$ during the overall process. If the adjuster can compensate at least its own compression during actuation, then $dx_{full} > 0$ which is minimum requirement. The stiffness during actuation is characterized

Table 4 Non-linear constraints for multi-objective optimisation

Nr.	Reason	Criteria	Case in Fig. 11
1	Flow model is inaccurate at high h/L	$h/L < 0.01$	
2	Flow model is inaccurate at high h/d_o	$h/d_o < 0.1$	
3	The adjuster is too flexible during actuation	$v_{act} < 0.1 \text{ mm/s}$	h
4	The adjuster is too rigid during adjusting	$v_{adj} > 0.001 \text{ mm/s}$	g
5	The adjuster works in Region 1		f
6	The adjuster works in Region 3		a

Fig. 12 Pareto front corresponding to different shear thickening fluids

by $dx_{\max} = x_1(t = 0 \text{ s}) - \min(x_1)$ maximal length change comparing to the initial state. The smaller $dx_{\max}$ means the stiffer adjuster during actuation. The design points are classified into five quality groups according to its $dx_{\max}$ and dx_{full} values:

- (A) Very good: Flexible and stiff, $dx_{\text{full}} > 0 \text{ mm}$, $dx_{\max} < 0.1 \text{ mm}$ (green)
- (B) Good: Flexible but moderately stiff, $dx_{\text{full}} > 0 \text{ mm}$, $0.1 \text{ mm} < dx_{\max} < 0.5 \text{ mm}$ (blue)
- (C) Acceptable: Flexible but not stiff, $dx_{\text{full}} > 0 \text{ mm}$, $dx_{\max} > 0.5 \text{ mm}$ (magenta)
- (D) Bad: Not flexible but moderately stiff, $dx_{\text{full}} < 0 \text{ mm}$, $0.1 \text{ mm} < dx_{\max} < 0.5 \text{ mm}$ (red)
- (E) Very bad: Not flexible nor stiff, $dx_{\text{full}} < 0 \text{ mm}$, $dx_{\max} > 0.5 \text{ mm}$ (black)

The color names in the brackets show the color of the corresponding markers and lines in Figs. 12 and 13. Figure 13 gives some example curves of the different quality groups to illustrate the classification rules (A)–(E).

The use of these quality groups and colors gives us the chance to investigate the Pareto fronts' population more carefully. *STF1*, *STF2* and *STF4* has no acceptable point at all. *STF6* has only bad points. That fluid is advantageous only in the case of longer recovery time because of the slow compression compensation.

On the other hand, both *STF3* and *STF5* have acceptable, good, and adequate points, too. According to the

Fig. 13 Typical examples of the different quality groups. The corresponding point is depicted in Fig. 12

functional requirements, both of them are suitable to use in an STF self-adjuster. Let us decide between these fluids from the financial point of view. *STF3* contains mono-disperse solid particles, which manufacturing is cumbersome; hence its price is high. In contrast to *STF3*, *STF5* is mixed from the commercial product of *Novotech* which results in almost the same functionality for a lower price. Consequently, *STF5* is recommended to use in STF-based self-adjuster.

4.2 Optimal geometry

The multi-objective optimization gives the geometrical dimensions of the dominant individuals. These dimensions are plotted against the first objective o_1 to make it easier to identify the corresponding point of Pareto front in Fig. 14. The results show that the D_d and D_u adjuster diameters are near constants during the whole domain, while dimensions of the flow restriction (h gap width, W gap circumference, L gap length) increases with the o_1 objective.

The very good individuals are in the domain of $o_1 < 0.1 \text{ mm}$, which implies a critical length, width and circumference. For *STF3* we have $L_{\text{crit}} = 140 \text{ mm}$, $h_{\text{crit}} = 18 \text{ mm}$ and $W_{\text{crit}} = 40 \text{ mm}$ and for *STF5* $L_{\text{crit}} = 90 \text{ mm}$, $h_{\text{crit}} = 25 \text{ mm}$ and $W_{\text{crit}} = 8 \text{ mm}$. These dimension combinations determine the gap geometry obviously. Below the critical values the adjuster would satisfy the requirements.

Fig. 14 Optimal geometrical dimensions corresponding to the points of the Pareto front in the case of *STF3* (right) and *STF5* (left)

4.3 Validation with CFD

The behavior of the individuals is now checked by transient time-dependent CFD simulations. Based on the solution of ODEs in each Pareto front, the p_u, p_d chamber pressures of the adjusters could be determined using Eqs. (9) and (10). The resulting $p_u(t)$ and $p_d(t)$ pressure(time) data pairs were used as inlet and outlet boundary condition of the unsteady CFD simulations in a look-up table format. The CFD simulations gave the $v(t)$ average flow velocity at the inlet as an output, which is identical to the adjuster's compression velocity. These velocities calculated by the simplified ODE-based method and the CFD technique are compared in Fig. 15. The CFD results show a slight delay compared to the ODE results in the actuation and releasing period. That effect can be explained by the fluid's inertia effect, which is not considered in the simplified ODE model.

The time integral of the compression velocity equals to actual length change of the adjuster. Therefore, the adjuster length is calculated from the CFD result as well. The comparison between the adjuster lengths corresponding to CFD and a simplified model is in Fig. 15. Although some time delay could be observed in the velocity(time) signal, these effects equalize each other. Hence, only an insignificant difference can be seen between CFD and simplified model results.

The CFD simulations are performed at each Pareto front point, giving the chance to recalculate the o_1 and o_2

objectives employing a more accurate CFD technique. The new $o_{1,CFD}$ and $o_{2,CFD}$ values are plotted on the previously presented Pareto graph, see Fig. 15 right hand side. The new Pareto front calculated by CFD moves only a slight to up in a high computational cost.

5 Discussion

5.1 Guideline to fluid choice

The investigated liquids have different material properties, e.g., (a) they have different $\mu(\dot{\gamma} = 1)$ zero viscosity, (b) different critical shear rate, where the shear thickening effect starts, and c) different shear thickening ability (the ratio of the minimum and maximum viscosity). There adjusting "performance" can be compared with its Pareto fronts depicted in Fig. 12. If we consider the rheological data in Fig. 6 as an input data, and the results in Fig. 12 as an output data, then the following statements can be taken:

- STF4 and STF6 has nearly the same zero viscosity, but STF4 gives worse Pareto points than STF6.
- Critical shear rates of STF1 and STF5 are close to each other, but STF5 produces adequate points compared to STF1. A similar situation arises in the case of STF6 and STF2.

Fig. 15 Pareto front (left hand side) corresponding to *STF5*, and the mechanical behavior corresponding to points of the different regions (right hand side)

- (c) The ratio of the highest and lowest viscosity values are 79.4, 31.6, 1671.2, 165, 344.4, 125.5 for STF1–STF6 respectively. If these fluids are sorted in increasing order based on their thickening ratios, we will get 1. STF3, 2. STF5, 3. STF6, 4. STF1, 5. STF2, 6. STF4 order. This order is the same as if they would be sorted based on the number of acceptable or adequate points on the Pareto fronts.

Statements (a)–(c) emphasize that only the fluid's thickening ratio has any significant effect on the adjuster behavior. It can be explained as follows.

There is only one unique $\tilde{Q}(\tau_p)$ flow curve for one prescribed rheology. If the zero viscosity is higher than the whole rheology moves upward, the flow curve would also move upward on a logarithmically scaled graph. On the other hand, if the critical shear rate increases, the flow curve moves to the left on a logarithmically scaled graph.

Constructing $\Delta p(Q)$ function from the $\tilde{Q}(\tau_p)$ flow curve means practically that all $\tilde{Q}$ points should be multiplied by Wh^2 (see Eq. (24)) and all τ_p value should be multiplied by

$\frac{L}{h}$. These operations would displace $\tilde{Q}(\tau_p)$ flow curve with $\log(Wh^2)$ in the x -direction and with $\log\left(\frac{L}{h}\right)$ in the y -direction logarithmically scaled graph, similarly to changing the critical shear rate or zero viscosity. As a result, the effect of the critical shear rate or the zero viscosity can be compensated by the proper geometrical dimensions. However, the shear thickening ability could not be improved by geometrical optimization.

Consequently, the more thickening fluid is advantageous to choose as adjuster liquid.

5.2 Validity of simplifications

Although the flow equations assume a quasi-steady process, the adjuster's practical operation is a dynamic mechanism. This steady-state assumption is checked via transient CFD simulations using the solutions of the optimization. The comparisons between the simplified model and CFD results in Fig. 15 show that the more accurate CFD results exhibit a slight delay compared to the simplified results. This delay detectable during the actuation and adjusting period as well, but they have no significant effect on the adjuster's behavior.

On the other hand, the Pareto points are recalculated via transient CFD simulations. The results in Fig. 15 present only a small discrepancy between the simplified and CFD results.

5.3 Application

Section 4 proves that our technique is applicable to design a shear thickening fluid-filled adjuster. The steps of the design are the following.

1. Chose a shear thickening fluid with a high thickening ratio.
2. Determine the $\tilde{Q}(\tau_p)$ relative flow curve using Eq. (24). For more details, see Nagy-György and Hős (2021) and Nagy-György and Hős (2019).
3. Check the accuracy of the $\tilde{Q}(\tau_p)$ relative flow curve via CFD simulations (Sect. 3.3).
4. Write governing equations according to Sect. 3.1. and substitute $\tilde{Q}(\tau_p)$ into them.
5. Perform multi-objective optimization on the geometrical dimensions of the adjuster using the previously defined governing equations.
6. Chose the final design considering practical aspects (manufacturing, built-in space, cost)
7. Test the behavior of the adjuster with transient CFD simulations.

6 Conclusions

A novel shear thickening fluid-based length compensating device was designed. The proposed adjuster is stiff during high compression load while flexible in the released state, which can be advantageous in, e.g., brake or clutch systems to compensate for the wear of brake/clutch disk. The adjuster consists of a cylinder in which a piston moves. The chambers between the cylinder's and piston's walls are filled with shear thickening fluid, which can flow between the chambers through an annular gap. The flow inside the annular gap and chambers is calculated by solving the simplified Navier-Stokes equation, while a system of ordinary equation systems models the behavior of the adjuster. Based on the coupled fluid-solid dynamic problem, a multi-objective optimization is performed on the adjuster's dimensions for six different shear thickening rheologies. The results give the optimal fluid and geometry for a particular application area. The simplified models' accuracy was checked by steady-state and transient CFD simulations using the rheological measurement data directly, without any model fitting. The proposed design method can be further developed to design other non-Newtonian fluid-based devices, e.g., dampers.

Acknowledgements Prepared with the professional support of the Doctoral Student Scholarship Program of the Co-operative Doctoral Program of the Ministry for Innovation and Technology from the source of the National Research, Development and Innovation Fund. The research reported in this paper was supported by the Higher Education Excellence Program of the Ministry of Human Capacities in the frame of Water science & Disaster Prevention research area of Budapest University of Technology and Economics (BME FIKP-VÍZ). The work of Péter Nagy-György was partially supported by Pro Progressio Foundation.

Funding Open access funding provided by Budapest University of Technology and Economics.

Declarations

Conflict of interest The authors declare that they have no conflict of interest.

Replication of results The necessary information for replication of the results is presented in this paper. The interested reader may contact the corresponding author for further implementation details.

Open Access This article is licensed under a Creative Commons Attribution 4.0 International License, which permits use, sharing, adaptation, distribution and reproduction in any medium or format, as long as you give appropriate credit to the original author(s) and the source, provide a link to the Creative Commons licence, and indicate if changes were made. The images or other third party material in this article are included in the article's Creative Commons licence, unless indicated otherwise in a credit line to the material. If material is not included in the article's Creative Commons licence and your intended use is not permitted by statutory regulation or exceeds the permitted use, you will

need to obtain permission directly from the copyright holder. To view a copy of this licence, visit <http://creativecommons.org/licenses/by/4.0/>.

References

- Amjad S, Shenbaga MN, Vinayaga RR (2007) Variable compression ratio engine: a future power plant for automobiles-an overview. *Proc Inst Mech Eng Part D J Autom Eng* 221(9):1159–1168
- Andrzej O, Sourav K (1995) A new method to solve generalized multicriteria optimization problems using the simple genetic algorithm. *Struct Optim* 10(2):94–99
- Ankita S, Abhijit M, Butola BS (2012) Improving the impact resistance of textile structures by using shear thickening fluids: a review. *Crit Rev Solid State Mater Sci* 37(2):115–129
- Bader J, Graf A, Heinzelmann K-F (2006) Releasing system with consistent stroke utilizing wear and tear compensation
- Barna S (2015) Observer based feedforward/feedback control of electro-pneumatic clutch systems. PhD thesis, (US20090181825A1). <https://repositorium.omikk.bme.hu/bitstream/handle/10890/1437/ertekezes.pdf?sequence=2> (visited: 2021-04-12)
- Bolognesi PW, Faria IS, Orsi CJ, Alexandre R (2017) Methodology of automatic slack brake adjuster definition considering foundation brake system characteristics. Technical report, SAE Technical Paper
- Burak G, Madhusudan R (2009) Simulation and test results for several variable-valve-actuation mechanisms. Technical report, SAE Technical Paper
- Chang KC, Yeh FY, Chen TW (2011) Feasibility study of shear-thickening fluid (STF) dampers. In: *Proceedings of 18th international conference on composite materials ICCM18*, Jeju Island, South Korea
- Chhabra RP, Richardson JF (2011) *Non-Newtonian flow and applied rheology: engineering applications*. Butterworth-Heinemann, Oxford
- Chien-Yuan H (2008) Fluid dynamics and behavior of nonlinear viscous fluid dampers. *J Struct Eng* 134(1):56–63
- Ciprian-Radu R, Olimpiu H (2017) An improved nonlinear modelling and identification methodology of a servo-pneumatic actuating system with complex internal design for high-accuracy motion control applications. *Simul Model Pract Theory* 75:29–47
- Conceição António CA (2013) Local and global pareto dominance applied to optimal design and material selection of composite structures. *Struct Multidisc Optim* 48(1):73–94
- Dac NV, Colin Navarrete Jose A, Hofman T, Steinbuch M, Serrarens A (2013) Optimal gear shift strategies for fuel economy and driveability. *Proc Inst Mech Eng Part D J Autom Eng* 227(10):1398–1413
- David F, Thomas H, Peter J (2016) Hydraulic lash adjuster compatible engine brake. *SAE Int J Engines* 9(4):2286–2291
- Dehghani S, Fathizadeh SF, Vosoughi AR, Noroozinejad FE, Yang TY, Hajirasouliha I (2020) Development of a novel cost-effective toggle-brace-curveddamper (TBCD) for mid-rise steel structures using multi-objective NSGA ii optimization technique. *Struct Multidisc Optim*. <https://doi.org/10.1021/acsami.1c06204>
- Dixon John C (2008) *The shock absorber handbook*, 2nd edn. Wiley, New York
- Eiben AE, Smith JE (2003) *Gray coding. Introduction to Evolutionary Computing*. Springer, New York
- Elizabeth M, María-Cristina R, Bertrand N (2014) A beginner's guide to tuning methods. *Appl Soft Comput* 17:39–51

- Eric B, Jaeger Heinrich M (2014) Shear thickening in concentrated suspensions: phenomenology, mechanisms and relations to jamming. *Rep Prog Phys* 77(4):046602
- Fang-Yao Y, Kuo-Chun C, Tsung-Wu C, Chung-Han Yu (2014) The dynamic performance of a shear thickening fluid viscous damper. *J Chin Inst Eng* 37(8):983–994
- Filip P, David J (2013) Applicability of the limiting cases for axial annular flow of power-law fluids. *Recent Adv Electr Eng Ser* 12:45–48
- Francesco C, Lorenzo S, Nicola B, Davide A (2017) Numerical and experimental investigation of a monotube hydraulic shock absorber. *Arch Appl Mech* 87(12):1929–1946
- Galindo-Rosales Francisco J (2016) Complex fluids in energy dissipating systems. *Appl Sci* 6(8):206
- Gibbs MS, Maier HR, Dandy GC (2010) Comparison of genetic algorithm parameter setting methods for chlorine injection optimization. *J Water Resour Plan Manage* 136(2):288–291
- Greenshields CJ (2017) Openfoam user guide. Version 6. OpenFOAM Foundation Ltd July
- Guo-Dong Z, Ji-Rong W, Long-Cheng T, Jia-Yun L, Guo-Qiao L, Ming-Qiang Z (2014) Rheological behaviors of fumed silica/low molecular weight hydroxyl silicone oil. *J Appl Polym Sci*. <https://doi.org/10.1002/app.40722>
- Hakan O, Yamin Jihad AA (2008) Performance and emission characteristics of lpg powered four stroke si engine under variable stroke length and compression ratio. *Energy Convers Manage* 49(5):1193–1201
- Hassan SM, Ehsan HS, Javad Z (2019) Cfd simulation and pareto-based multi-objective shape optimization of the centrifugal pump inducer applying gmdh neural network, modified nsga-ii, and topsis. *Struct Multidisc Optim* 60(4):1509–1525
- Hong SR, Wereley NM, Choi YT, Choi SB (2008) Analytical and experimental validation of a nondimensional Bingham model for mixed-mode magnetorheological dampers. *J Sound Vib* 312(3):399–417
- Hong Z, Lixun Y, Wanquan J, Shouhu X, Xinglong G (2016) Shear thickening fluid-based energy-free damper: design and dynamic characteristics. *J Intell Mater Syst Struct* 27(2):208–220
- Hongxing G, Maoru C, Liangcheng D, Jungang Y, Xiaozhi Z (2020) Mathematical modelling and computational simulation of the hydraulic damper during the orifice-working stage for railway vehicles. *Math Probl Eng*. <https://doi.org/10.1155/2020/1830150>
- Jafaryeganeh H, Ventura M, Guedes Soares C (2020) Effect of normalization techniques in multi-criteria decision making methods for the design of ship internal layout from a pareto optimal set. *Struct Multidisc Optim* 62:1849–1863
- Jian S, Frazer John H, Tang M (2007) Shape optimisation using evolutionary techniques in product design. *Comput Ind Eng* 53(2):200–205
- Kalyanmoy D (2001) Multi-objective optimization using evolutionary algorithms, vol 16. Wiley, New York
- Kalyanmoy D, Amrit P, Sameer A, Meyarivan T (2002) A fast and elitist multiobjective genetic algorithm: NSGA-II. *J Mag* 6(2):182–97
- Kambiz E, Ashutosh B, Ramin S (2014) Dynamic analysis of electro- and magneto-rheological fluid dampers using duct flow models. *Smart Mater Struct* 23(3):035016
- Kun L, Hongjun L, Minghai W, Annan Z, Fanrui B (2018) Dynamic performance of shear-thickening fluid damper under long-term cyclic loads. *Smart Mater Struct* 28(2):025007
- Kyongsol K, Zhaobo C, Dong Yu, Changhyon R (2016) Design and experiments of a novel magneto-rheological damper featuring bifold flow mode. *Smart Mater Struct* 25(7):075004
- Leyla F, Mohamed D, Khaled Z, Younes S (2019) Adaptive sliding mode vibrations control for civil engineering earthquake excited structures. *Int J Dynam Control* 7(3):955–965
- Liang-Liang S, Dang-Sheng X, Cai-Yun X (2013) Application of shear thickening fluid in ultra high molecular weight polyethylene fabric. *J Appl Polym Sci* 129(4):1922–1928
- Ljøsne, KT (2015) Automatically adjusting clutch actuator
- Marcin O, Rifat K, Marcello O, Nicola T (2008) Application of an integrated valvetrain and hydraulic model to characterization and retuning of exhaust valve behavior with a dpf. Technical report, SAE Technical Paper
- Marler RT, Arora JS (2004) Survey of multi-objective optimization methods. *Struct Multidisc Optim* 26:369–395
- Máté B, Péter C, Sára T (2020) Experimental and numerical investigation of the loss coefficient of a 90 pipe bend for power-law fluid. *Period Polytechn Chem Eng* 64(4):469–478
- Mathieu S, Véronique M (2014) Energy dissipation in concentrated monodisperse colloidal suspensions of silica particles in polyethylene glycol. *Colloid Polym Sci* 292(12):3291–3299
- Miad ZS, Pourtakdoust Seid H (2018) Multidisciplinary design of a guided flying vehicle using simplex nondominated sorting genetic algorithm ii. *Struct Multidisc Optim* 57(2):705–720
- Miao Yu, Xiuying Q, Xingjian D, Kang S (2018) Shear thickening effect of the suspensions of silica nanoparticles in peg with different particle size, concentration, and shear. *Colloid Polym Sci* 296(7):1119–1126
- Minghai W, Kun L, Qian G (2018) Modeling mechanical properties of a shear thickening fluid damper based on phase transition theory. *EPL (Europhys Lett)* 121(5):50001
- Minghai W, Gang H, Lixiao L, Haitao L (2018) Development and theoretically evaluation of an stf-sf isolator for seismic protection of structures. *Meccanica* 53(4–5):841–856
- Minghai W, Kun L, Hongjun L (2019) Experimental investigation on hysteretic behavior of a shear thickening fluid damper. *Struct Control Health Monit*. <https://doi.org/10.1002/stc.2389>
- Mitchell M (1998) An introduction to genetic algorithms, isbn 9780262631853
- Mladenko K (2019) Unconstrained shape optimisation of a lightweight side door reinforcing crossbar for passenger vehicles using a comparative evaluation method. *Int J Automot Technol* 20(1):157–168
- Muaz K, Tahsin E (2021) Numerical analysis of a monotube mixed mode magnetorheological damper by using a new rheological approach in cfd. *Rheol Acta* 60(1):77–95
- Muralidharan K, Vasudevan D (2011) Performance, emission and combustion characteristics of a variable compression ratio engine using methyl esters of waste cooking oil and diesel blends. *Appl Energy* 88(11):3959–3968
- Nenno Paul T, Wetzel Eric D (2014) Design and properties of a rate-dependent ‘dynamic ligament’ containing shear thickening fluid. *Smart Mater Struct* 23(12):125019
- Nicola G, Andrea S (2017) Experimental validation of a novel magnetorheological damper with an internal pressure control. *J Intell Mater Syst Struct* 28(18):2489–2499
- Nunney MJ (2007) Light and heavy vehicle technology. Routledge
- Péter C, Csaba H (2013) Predicting the friction factor in straight pipes in the case of Bingham plastic and the power-law fluids by means of measurements and CFD simulation. *Periodica Polytech Chem Eng* 57(1–2):79–83
- Péter C, Csaba H (2014) CFD-based estimation and experiments on the loss coefficient for Bingham and power-law fluids through diffusers and elbows. *Comput Fluids* 99:116–123
- Péter N-G, Csaba H (2019) A graphical technique for solving the Couette–Poiseuille problem for generalized Newtonian fluids. *Periodica Polytech Chem Eng* 63(1):200–209
- Péter N-G, Csaba H (2021) Predicting the damping characteristics of vibration dampers employing generalized shear thickening fluids. *J Sound Vib* 506:116116

- Péter C, Sáira T (2018) The effect of rheology model of an activated sludge on the predicted losses by an elbow. *Period Polytechn Mech Eng* 62(4):305–311
- Pradeep C (2008) Design of automatic slack adjuster for drum brake in 150cc 4stroke bike on indian road. Technical report, SAE Technical Paper
- Qian Z, Yonghui H, Hongliang Y, Bangchun W (2018) Dynamic performance and mechanical model analysis of a shear thickening fluid damper. *Smart Mater Struct* 27(7):075021
- Qianyun H, Xinglong G, Shouhu X, Wanquan J, Qian C (2015) Shear thickening of suspensions of porous silica nanoparticles. *J Mater Sci* 50(18):6041–6049
- Roberts L, Jr JMC (2018) Design and development of a roller follower hydraulic lash adjuster to eliminate lash adjustment and reduce noise in a serial production diesel engine. Technical report, SAE Technical Paper
- Sean B (2015) Heavy duty truck systems. Cengage Learning, Boston
- Selim G, Cemal KM, Weihua L (2017) Shear thickening fluids in protective applications: a review. *Prog Polym Sci* 75:48–72
- Serrano-Aguilera JJ, Parras L, del Pino C, Rubio-Hernandez FJ (2016) Rheo-piv of aerosil® r816/polypropylene glycol suspensions. *J Nonnewton Fluid Mech* 232:22–32
- Shellhause RL (1987) Drum brake adjusters, November 17. US Patent 4,706,784
- Shwetank K, Syahrir R, Pandian V, Umer IS, Ntow OT (2020) Simplified predictive model for downhole pressure surges during tripping operations using power law drilling fluids. *J Energy Resour Technol* 142(12):33–35
- Snyder Rebecca A, Kamath Gopalakrishna M, Wereley Norman M (2001) Characterization and analysis of magnetorheological damper behavior under sinusoidal loading. *AIAA J* 39(7):1240–1253
- Solberg HH (2011) Self-adjusting clutch actuator for operating a vehicle clutch
- Sujuan J, Jiajin T, Hui Z, Hongxing H (2017) Modeling of a hydraulic damper with shear thinning fluid for damping mechanism analysis. *J Vib Control* 23(20):3365–3376
- Sultan YB, Hüseyin L (2017) Fatigue-based structural optimisation of vehicle components. *Int J Veh Des* 73(1–3):54–62
- The Mathworks, Inc., Natick (2017) Massachusetts. MATLAB version 9.3.0.713579 (R2017b)
- Tian Xu, Jie Li (2020) Robust aerodynamic shape optimization using a novel multi-objective evolutionary algorithm coupled with surrogate model. *Struct Multidisc Optim* 62:1969–1987
- Tongfei T, Masami N (2017) Design and testing of a rotational brake with shear thickening fluids. *Smart Mater Struct* 26(3):035038
- Tudon-Martinez Juan C, Diana H-A, Luis A-B, Ruben M-M, Jorge L-S, de Aquines Osvaldo, J (2019) Magneto-rheological dampers—model influence on the semi-active suspension performance. *Smart Mater Struct* 28(10):105030
- Unsanhame M, Abhijit M, Animesh L (2020) A review of fibrous materials for soft body armour applications. *RSC Adv* 10(2):1066–1086
- Weifeng J, Guangjian P, Yi M, Heng C, Jiangjiang H, Chao J, Taihua Z (2017) Measuring the mechanical responses of a jammed discontinuous shear-thickening fluid. *Appl Phys Lett* 111(20):201906
- Woo YJ, Francesca R, Theophane C (2019) Road noise reduction of a sport utility vehicle via panel shape and damper optimization on the floor using genetic algorithm. *Int J Automot Technol* 20(5):1043–1050
- Woon Soon Y, Querin Osvaldo M, Steven Grant P (2001) Structural application of a shape optimization method based on a genetic algorithm. *Struct Multidisc Optim* 22(1):57–64
- Xueyi W, Lianbo M, Shujun Y, Min H, Xingwei W, Junfei Z, Xiaolong S (2020) An aggregated pairwise comparison-based evolutionary algorithm for multi-objective and many-objective optimization. *Appl Soft Comput* 96:106641
- Yuki S, Nagao Y, Motoyasu S, Kenta Y (2013) Dynamic characteristic calibration of a hydraulic lash adjuster model using unit excitation test. Technical report, SAE Technical Paper
- Zhang XZ, Li WH, Gong XL (2008) The rheology of shear thickening fluid (STF) and the dynamic performance of an STF-filled damper. *Smart Mater Struct* 17(3):035027

Publisher's Note Springer Nature remains neutral with regard to jurisdictional claims in published maps and institutional affiliations.