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Towards a modelling framework for designing active check valves – a review of state-of-the-art

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ABSTRACT

This paper proposes a design methodology for designing the mechanical topology in hydraulic active check valves (ACVs). These valves may have optimised flow geometries to obtain fast switching and high flow rates. Furthermore, ACVs comprise large pressure differentials and high deceleration during plunger and seat impact, which introduces a concern about the severity of long-term fatigue and wear. A modelling framework of ACVs with the desired capabilities is multidisciplinary involving analysis of mechanics, fluid dynamics, contact mechanics and material science. Therefore, a state-of-the-art focusing on the significant contributions of these disciplines has been conducted. The literature review is focusing on lifetime models and wear concerned with valve seat inserts in different fatigue cycle regimes. No studies with direct application have been found, making this work an infant step towards lifetime prediction of ACVs. However, studies of similar systems have pinpointed the mechanisms that result in wear particles. It is the main objective to uncover the parameters with greatest significance when considering lifetime estimation of ACVs and a conceptual formulation of a lifetime framework is established.

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1. Introduction

A new generation of fluid power technology, digital hydraulics, is getting increased attention (Ehsan *et al.* 1997, Payne *et al.* 2005, Roemer 2014, Brandstetter *et al.* 2016). This technology is used in hydraulic motors/pumps, which is attractive due to its potential to increase part load efficiency significantly when compared to traditional hydraulic machines. A conceptual schematic of such a system is shown in Figure 1 where the pistons are interacting with an eccentric drive that by rotation causes compression and expansion of the piston chambers. By moving fluid from a chamber through active check valves (ACVs) to a high- or low-pressure manifold, the system may be controlled to meet the application requirements. For example, an interconnection of a digital displacement pump and a DD motor (DDM) may replace the mechanical transmission system in a wind turbine, with the objective of enhancing scalability. However, the reliability of DDMs is yet to be validated and the only current running prototype of a DD transmission in wind turbines is by Mitsubishi Heavy Industries (Mitsubishi Heavy Industries Ltd 2015) and is monitored constantly. Among others, the reliability is limited by the lifetime of the ACVs.

A well proven approach to design hydraulic ACVs does not exist at the current state of research, nor have the significant wear mechanisms been identified. Therefore, a flowchart of the ACV design has been established as seen in Figure 1 inspired by Roemer *et al.* (2012). As seen in the figure, the modelling framework is what is of interest in this paper. A major task is here to evaluate the operating conditions (temperatures, local pressures, contact-impact speeds, loads, fluid flow, etc.) of an ACV applied in some arbitrary system, e.g. a DDM. To accomplish this, the design tool needs to include multiple physical phenomena such as fluid dynamics, structural mechanics and thermohydrodynamics. The review emphasises current state-of-the-art within each area, thus only focusing of main contributions. The objective is to give an overview of the design problem by functional decomposition, hereby highlighting the required modelling. The ACV design will among other factors be affected by wear. Studies of wear have shown that universal wear models are not yet identified (Hsu *et al.* 1997, Williams 1999, Van 2013). Thus, empirical data from ACVs is essential to iterate towards an applicable modelling framework. This is the basis for the current paper which is organised as follows: A state-of-the-art of ACVs is presented in Section 2 to show state-of-the-art design approaches. In Section 3, the structure of the modelling framework is illustrated and the concept elaborated. The modelling framework requires; computational fluid dynamics (CFD), contact mechanics including numerical methods suitable for design of

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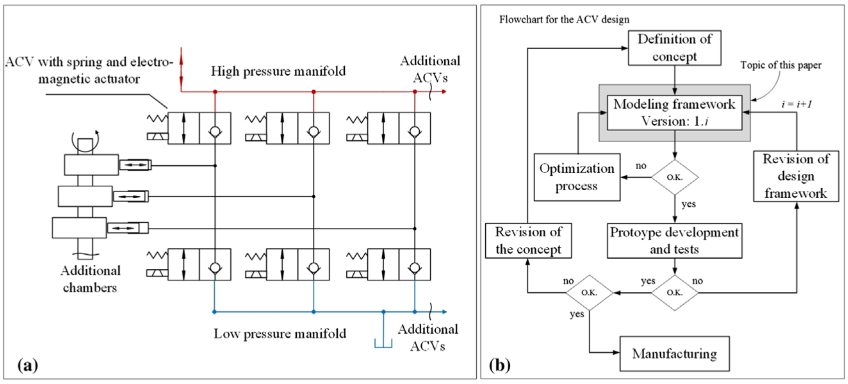

Figure 1. (a) Schematic of a DD pump/motor with high pressure valve (HPV) and low pressure valve (LPV) with inspiration from (Noergaard *et al.* 2016a). (b) Flowchart for the ACV design derived from a combination of experimental and a Computation Aided Engineering-based design process (Kaltenbacher 2007).

ACVs; and modelling of fatigue and wear phenomena. These areas are reviewed through Sections 4–6 and the conclusions are given in Section 7.

2. State-of-the-art for ACVs in digital fluid power

The incentive for the proposed design methodology is twofold; firstly, the fluid dynamic properties of the valve are required for design purposes; secondly, the durability of the valves is a major concern for the industrial partners developing DD machines. The technology is in its infant phase and there does not exist any official data specifically for DD machines. However, Paloniitty *et al.* (2016) have tested durability of water hydraulic on/off valves with 10^6 cycles. The authors aim to study durability of different coatings and hardening techniques. Furthermore, an elastic plunger design was proposed, but test results did not show increase of durability by that design. The tests were performed with a pressure level of 14 MPa at 80 Hz switching frequency, which is therefore comparable to the situation in a DDM. All valve specimens show significant wear around both seat and plunger, the depth of wear on the plunger range between 0.01 and 0.06 mm. Water hydraulics is prone to wear when compared with oil hydraulics, but this finding indicates that durability models are relevant along with design topologies that can limit wear.

2.1. Existing design methodologies

A detailed study of digital hydraulic valves and definition of 'the perfect', yet impossible, valve is given in (Linjama and Vilenius 2008), where relevant points from the definition reads: 'infinite bandwidth, unlimited durability and characteristics independent on fluid, temperature, pressure, wear etc.' As an example, specifications for ACVs for DDM operation include switching times around 3 ms at a rated speed of 800 rpm, along with high flow rates (>120 l/min) at pressure drops of ≈0.5 bar (Noergaard *et al.* 2016b). Furthermore, the required lifetime of DD machines is up to 25 years. This potentially results in $>10^{10}$ switching cycles of the ACVs, introducing a situation where classical fatigue analysis design approaches may not be sufficient.

State-of-the-art in design of ACVs is presented in (Roemer *et al.* 2012, 2013a, 2014a) where different topics of the design phase are addressed, e.g. actuation and flow geometry. The proposed approach for ensuring the durability of the design is to evaluate the modified-Mohr effective stress and compare that with the fatigue strength under tensile load of the selected material with a safety factor of 1.5. For any contact problem, multiple factors will influence the durability. However, the authors do state that the lifetime of the ACV may be limited by local wear phenomena. Therefore, it is also noted that it may not be possible to accurately determine the lifetime from the presented approach. This problem is thus proposed for further research.

Several suggestions and attempts have been presented for design of digital valves with the required capabilities, e.g. (Winkler and Scheidl 2007, Mahrenholz and Lumkes 2009, Plöckinger *et al.* 2009b, Uusitalo *et al.* 2010, Winkler *et al.* 2010, Linjama *et al.* 2015). All designs are on/off types and serve the same purpose, but are in general differing in choice of actuation topology and flow geometry. However, one of the commonalities of the design procedures is a lack of durability considerations. The requirement is mentioned in correlation with the piloted multi-poppet valve presented by Winkler *et al.* (2010). The same valve is tested with 10^8 cycles without any issues presented by Plöckinger *et al.* (2009a). This approach with physical prototyping and afterwards durability testing is not optimal from a design perspective. However, it may be necessary with the current state of lifetime prediction models. This is one of the topics investigated in this paper.

3. Suggested modelling framework

The objective of this paper is to define a mathematical modelling framework for designing ACVs. This is done based on the flowchart structure presented in Figure 2 and an overview of state-of-the-art in the various areas.

The framework is divided into five main parts, each part is sequentially depending upon the prior part, and iterative steps between the parts are expected when defining the appropriate model complexity. The overall idea is that numerical models can be used to analyse a given topology and from this simplified models for optimisation purposes are pursued. The reasoning behind each part is: Part I involves interaction with the design engineer to obtain the requirements for the valve design. This takes the defined information and sends it to appropriate analysis tools. In Part II the information is used to establish CAD models with discretisation of the geometry. This is necessary for the computations of the fluid dynamic properties and material stresses in part III and IV. The computation of local fluid pressure and fluid velocity profiles in part III are used to obtain the impact velocity between valve plunger and seat along with the flow characteristics and valve switching time. Numerical analyses may require iterative steps, which is why iterative changes to the solver settings are allowed. Furthermore, the CFD analysis is dynamic. Therefore, changes in meshing techniques of the fluid domain are possible which may require changes in the CAD as well. The computed fluid pressures and dynamics of the plunger are used in part IV for a structural analysis of the material stresses, which combined are used to derive relevant lumped models of the investigated topology, e.g. flow, flow forces and Hertzian stress. Part V uses the derived models to evaluate the objective functions of the given topology and returns the parameters relevant for the given analysis, some examples are shown in the Figure 2. The numerical methods required through part III & IV are a cornerstone of the framework and reviews are given in the following two sections.

4. Computational fluid dynamics

The objective of the CFD analyses is to calculate fluid pressure and fluid velocity profiles (part III in Figure 2). The dynamics of the plunger moving through the fluid may be estimated with a transient numerical analysis of the fluid pressures. Results from the numerical analysis may be used to form expressions for the movement-induced forces on the plunger and for the flow properties of the design. Specifically, the fluid dynamics associated with ACVs are considered as a rigid body moving through a hydraulic oil. The assumptions regarding rigidity of the armature and seat are only an issue at very small gap heights (≤50 μm) since the pressure of a thin film is highly dependent of this height. Any deformation would therefore result in solution inaccuracies. However, for the most part of the fluid domain, the solution is not significantly affected by assuming a rigid body. The force equilibrium for the plunger is described by:

$$m\ddot{z} = F_{\mathrm{act}}(z, i) + F_f(z, Q, p, t) + F_s(z) \tag{1}$$

where m is the mass of the moving member, i is the current supplied to the actuator coil, F_{act} is the actuator force, F_f is the force related to interaction of the moving member and the surrounding fluid, F_s is the spring force and z is the armature displacement defined as seen in Figure 3. As mentioned, F_f is the objective of the CFD analysis combined with lumped parameter models.

4.1. General CFD state-of-the-art related to ACVs

CFD results depend on geometry, defined mesh and solver settings, and in general there is no universal solution strategy. However, studies of CFD on applications similar to the ACV give an indication of how to solve the problem.

Individually, the shape of both HPV and LPV are axisymmetric with the exception of some venting holes, and it will simplify the analysis significantly if a 2D CFD analysis is sufficient. The validity of such analysis need to be investigated for the specific application either

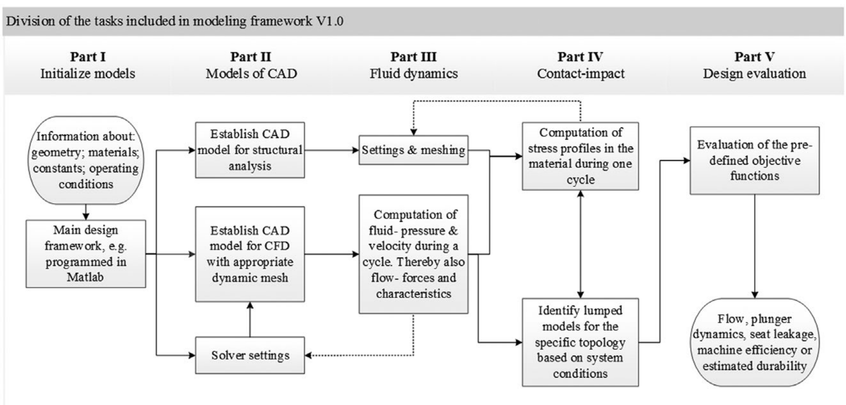

Figure 2. The flowchart of the suggested modelling framework for designing the mechanical topology of ACVs.

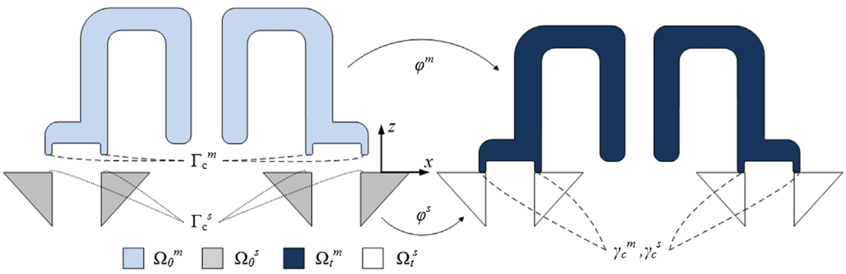

Figure 3. Schematic of a 2D cross section of the valve seat insert (VSI) in the prototype valve with the space before and during contact where contact will involve a certain deformation of the original bodies.
Note: Symbols are following standard mortar formulation, and φ describes maps between spaces.

mathematically, by full 3D analysis or experimentally. Chattopadhyay *et al.* (2012) document an example of this approach with a static analysis of a variable spool-type orifice.

Simic and Herakovic (2015) develop a 3D CFD analysis with the objective of optimising a seat valve by minimising the flow forces. It is shown that the precision of turbulent models for the flow force calculation via the shear-stress-transport (SST) k-ω model is superior compared to the k-ε and k-ω models. In general, a Reynolds Averaging of the Navier-Stokes equations solved by the linear Eddy viscosity two-equation models is the industry standard and used for many different engineering problems. By means of averaging the pressure and velocity fields, the computational time may be drastically reduced (this approach is invalid in rotating flows or if flows are heavily curved, which in the literature has shown not to be an issue for hydraulic valves). The foundation of the two-equation models, i.e. the Boussinesq Eddy viscosity assumption, is not valid in the viscous sub-layer near solid walls, which is why additional models (low-Reynolds number models) have been developed to overcome this issue. The desired analysis includes analysis of flow in the near wall region where both accurate flow forces and pressures in wall regions are important to describe the dynamic performance of the ACV. Simic and Herakovic (2015) addresses meshing in the near-wall region and it is shown to have a significant influence on the solution.

4.2. CFD on ACVs

Recent advances in CFD applied to ACVs are presented by Roemer *et al.* (2012, 2013b, 2014a, 2015b) where transient CFD routines accommodating moving boundaries are used to analyse valve opening and closing. In (Roemer *et al.* 2012), a design process of a ACV is suggested, where a key component is CFD and in particular a steady 3D analysis is performed to investigate the flow characteristics, which is compared with measurements and a lumped parameter model. This is expanded in (Roemer *et al.* 2013b), where the work comprises transient 3D analysis applied for ACVs in a DD unit. The calculated pressures are used to evaluate the DD unit's efficiency. The computational effort of this model makes it insufficient for use in optimisation. Therefore, a transient 2D axisymmetric analysis is performed in (Roemer *et al.* 2014a), and this is used to derive an expression for the movement-induced forces, which is more feasible when formulating the design problem. However, the effort concerned with this simulation is not comparable with analytical approximations models. The 2D model is solved by the Re-Normalisation Group k-ε two-equation turbulence model, which accounts for history effect-like convection and diffusion of turbulent energy. Hereby, it is able to account for the 'history' of turbulence, making it appropriate for problems with moving boundaries and for flow profiles with a transition from laminar to turbulent flow conditions. As previously discussed, two-equation models must be modified with low-Reynolds models for analysis of the viscous sub-layer. Therefore, this was applied to simulate the valve.

Common for the above references is that only small parts of the computations are validated and the tribological thin film effects during the closing event are not analysed in detail either. The reviewed literature of CFD applied for ACVs assumes rigid body motion, and no deformation is included in the CFD formulation which may influence the pressure field when plunger and seat are near contact. On the contrary, most of the work done in tribology for fluid power applications (e.g. axial piston pumps) underlines the need of using deformation when simulating thin film or mixed lubrication problems (Pelosi and Ivantysynova 2012, Cerimagic *et al.* 2016, Chacon and Ivantysynova 2016), which will occur in the ACV. Also, the fluid viscosity temperature and pressure dependency is reported to influence the solution of the pressure field significantly. The assumption of rigid body motion is therefore only considered valid to determine the fluid dynamics when the plunger is not near the seat (above 50 μm).

4.3. Lumped parameter models in fluid dynamics

Simplifications of the computational heavy numerical solution by analytical approximations are a common approach. For valves, this approach typically involves use of the orifice equation along with the continuity equation applied for desired control volumes. Furthermore, oil stiffness models and a modified discharge coefficient depending on Reynold's number can be included. Furthermore, the forces from fluid interaction as included in the CFD simulation may be represented by an analytic expression accounting for flow forces and movement-induced forces including the history effects as presented in (Borutzky *et al.* 2002) and formulated as:

$$F_f(z, Q, t) = F_{\text{flow}}(z, Q, p) + F_{\text{mov}}(z, t) \tag{2}$$

where F_{flow} is the force induced by the flow as seen in Equation (3). This expression includes both turbulent and laminar flow coefficients and F_{mov} is the movement-induced force arising by rigid body motion through an otherwise still, incompressible and viscous fluid (Lai and Mockros 1972). This force can be approximated with Equation (4), which is an expression derived from a linearised form of the Navier-Stokes equations.

$$F_{\text{flow}}(z, Q, p) = \begin{cases} K_{f,1}(z)Q^2 + K_{f,2}(z)Q, \text{ if } z > 0 \\ A_p \Delta p, \text{ if } z = 0 \end{cases} \tag{3}$$

$$F_{\text{mov}}(z, t) = K_a(z)\ddot{z} + K_v(z)\dot{z} + K_d\dot{z}|\dot{z}| + K_h \int_0^t \frac{\frac{\mathrm{d}\dot{z}}{\mathrm{d}\tau}}{\sqrt{t-\tau}} \mathrm{d}\tau \tag{4}$$

where A_p is the flow passage area, K_a is the virtual mass of accelerated fluid, K_v is the viscous shearing coefficient, K_d is the drag coefficient, $K_{f,i}$ is the flow force coefficients, K_h is the history coefficient and τ is the step size. The time dependency represents the historical effect, which may either be determined via CFD analysis or estimated by an analytic expression. The exact formulation and methods of derivation of the fluid coefficients related to F_f is left out, but can be found in (Roemer *et al.* 2015b).

4.3.1. Stiction and squeeze film effects

The early research concerning stiction is done in the field of automatic compressor valves. This includes experimental evaluation by Giacomelli and Giorgetti (1974), and by means of simulation by Bauer (1990) using viscous models, and recently using Reynolds equation to solve Navier-Stokes equations by Pizarro-Recabarren *et al.* (2013). None of the aforementioned references take into account fluid cavitation, although studies of separation of two plates have shown that cavitation and fingering effects limit the stiction force (Budgett 1911, Poivet *et al.* 2003, 2004, Lindner *et al.* 2005).

A seat valve's ability to open within short periods of time entails quick separation of two contact surfaces. Furthermore, a liquid may be present between the surface which may give rise to a significant force opposing opening movement (Roemer *et al.* 2015a). This phenomenon is referred to as stiction and is influenced by the geometry of the contact surfaces, and may be a cause of wear in the valve seat insert (VSI) due to tensile stress and cavitation in the thin fluid film during the separation of two contact surfaces. The squeezed film effect which affects the impact velocity gives rise to a dampening effect. A low impact velocity will result in lower peak stress and is thus important to understand the impact wear mechanism of the VSI. Therefore, the approaches for modelling of this are of major importance.

The approach of utilising Reynolds theory has most recently been investigated in (Resch and Scheidl 2013, Scheidl and Gradl 2013, 2016, Roemer *et al.* 2014b, 2015a) where (Resch and Scheidl 2013, Scheidl and Gradl 2013) proposes a stiction model with a central cavitation zone with constant zero pressure, where liquid tension is not allowed. Furthermore, surface tension is neglected in the cavitation zone due to its demonstrated insignificance. When cavitation is present, the pressure of the non-cavitated fluid is described by the Reynolds equation and the boundaries of the cavitation zone are solved dynamically with a time-stepping solver, whereby the zone can expand and collapse. This approach gives good estimation of the stiction force at relatively large plate distance, but very small distances (below 0.2 mm) makes the assumption of no tensile strength of the liquid invalid. Thus, a larger force will occur in reality when compared to this model. In (Roemer *et al.* 2014b, 2015b), a stiction model based on three different situations with different boundary conditions is analysed. This model allows for a fixed tensile strength of the liquid and the occurrence of negative pressures. The exact tensile strength depends on purity and the smoothness of the surrounding surfaces. Scheidl and Gradl (2016) present an approach to approximate the stiction force, hence reducing the required computational effort. The approach requires a finite initial gap height, i.e. mechanical contact cannot be present. The stiction is evaluated by solving a two-dimensional Poisson problem and the situation of an expanding cavitation zone is thereby given analytically. While the cavitation zone is shrinking, two non-linear ODE's are solved to determine the cavitation domain.

Studies of squeeze film damping (Keating and Ho 2001, Nayfeh and Younis 2004, Scheidl *et al.* 2014, Scheidl and Gradl 2015) also relies on the Reynolds equation to solve Navier-Stokes equations. Examples of analytic squeeze film damping and stiction are given in (Scheidl *et al.* 2014, Roemer *et al.* 2015b) where it is assumed that; the process is quasi-static, fluid viscosity is constant and gap height (z, see Figure 3) is significantly smaller than the surface length (L), e.g. $z \leq 0.1L$. From the Reynold's equation, the pressure distribution in the gap thus reduces to:

$$p(x) = \frac{6\mu\dot{z}x^2}{z^3} + C_1x + C_2 \tag{5}$$

where μ is the dynamic viscosity of fluid, C_i are integration constants and x is the radial direction (see

Figure 3 for reference). By assuming uniform pressure in Cartesian coordinates around the z-axis, $p(x)$ may be integrated over x and then over $[0–2\pi]$ to obtain the pressure acting on the shadow area. In this manner, the force from both fluid stiction and squeeze film may be approximated. However, as shown in (Roemer *et al.* 2015b) more accurate boundary conditions may lead to lower and more accurate forces and cavitation of the fluid will result in lower stiction force. Finally, effects caused by a pressure and temperature-dependent viscosity may contribute to significant deviations from the pressure approximated under assumption of constant viscosity. The piezo-viscous squeeze film is studied by Johansen *et al.* (2017) for one torus approaching a fixed torus with some initial velocity before end-damping. This geometry is similar to the annulus shape of the ACV, and the piezo-viscous model predicts that local pressures will be higher than when the contact problem is considered dry. From a durability perspective, this conclusion is interesting but the consequence of such local pressures is not yet clarified.

In relation to this, a design proposition towards reducing the impact forces is presented by Scheidl *et al.* (2014) and Scheidl and Gradl (2015) addressing both fluid stiction and squeeze film effects for hydraulic valves with the objective of reducing the impact wear while decreasing the stiction force. Based on this, a cushioning-type design is presented by which an enhanced dampening effect is achieved. The theoretical study is based on some fluid dynamical simplifications from using the orifice equation, uniform contact and rigid bodies. This is where a numerical analysis can aid in understanding the system, which is why this is essential to the design methodology.

5. Simulation of contact mechanics

The plunger and seat may be squeezed against each other by a large pressure difference, resulting in metal on metal contact. Furthermore, the kinetic energy of the armature is converted upon breaking, potentially causing stress. This combination is from now on referred to as contact-impact, which is represented by part IV from Figure 2. Contact-impact problems are highly non-linear and non-smooth due to geometrical, material and contact nonlinearities. The discontinuous contact constraints allows sudden transmission of forces from one part of the model to another making it difficult to solve, which is why appropriate algorithms are necessary. Contact of the two bodies of interest in an ACV mainly causes compressive–compressive mechanical stress in both bodies. A purely compressive stress is normally not associated with fatigue, but in reality, there will always be some level of tensile stress, and subsurface cracks will nucleate over time (Van 2013, Ch. 6). The contact stress may be approximated analytically for simple geometries, e.g. by the Hertzian stress as explained in Section 6.1.5, thus reducing the simulation effort. However, if friction, deformations and complex geometries are part of the problem it is convenient to apply numerical tools such as the finite element method (FEM) to accurately calculate stresses (still requires knowledge about the local friction). The Hertzian solution is interesting to be used as a lumped model for ACVs, but this must initially be supported by FEA.

One of the first attempts using FEM on contact problems was performed by Francawilla and Zienkiewicz (1975), and a year later Hughes *et al.* (1976) presented a detailed study on the topic. In (Francawilla and Zienkiewicz 1975), it is shown that the FEM solution is not only superior to the Hertzian solution, but also that quite complex problems may be solved inexpensively with FEM. The FEM reduces a complex object to simpler elementary parts. The contact problem then involves both detecting and describing how the discretised bodies interact during contact. For example, if deformation causes a change in contact status of different elements or if tangential friction forces are present. By defining appropriate constraints for the specific problem, the material pressures may be approximated by numerical analysis for the domain of interest. For details about computation of contact-impact problems the reader is referred to the monograph on the subject by Laursen (2003). Furthermore, a detailed description of solid surface contact is given by Williams and Dwyer-Joyce (2001, Ch. 3), where among other the process of elastic shakedown is explained. This strengthens the materials if solid on solid contact occurs with entirely elastic deformation. Increase of 50% yield strength is reported. Such contact is present in ACVs and is interesting when considering the fatigue limit of the ACVs.

The ACV contact problem is illustrated by Figure 3 showing parts of the two bodies with one before contact (Ω_0^i) and another during contact (Ω_t^i). This accounts for possible deformations changing the original set and allows reformulation of the mesh. Furthermore, the surface areas which are relevant for the contact problem (Γ_c^i) are used to update boundary conditions and contact constraints, which allow the global mechanical system to be analysed in a local setting. This formulation is thus used to join the two contacting surfaces as explained in (Laursen 2003), which is beneficial for the convergence of the solution.

5.1. Considerations for static analyses

The ACV contact-impact problem consist of mixed-lubrication, where the contacting surfaces (Γ_c^i) are not changed with time, as would be the case in, e.g. rotational, sliding or rolling simulation problems. This means that the surfaces in contact are well-known beforehand. However, the interaction of asperities on the two surfaces and the influence of the presence of a lubricant are what makes this a computationally difficult contact-problem. There exist a number of suitable formulations to be used for contact mechanics and most formulations are derived to be suited for one special case, i.e. contact-impact, tangential sliding, sharp contact, large transformations, e.g. when bending material. The following conditions are considered to be valid and relevant for static analysis of an ACV (Hughes *et al.* 1976):

- impenetrability,
- the intersection of material points of the surface boundaries is at some point in time different from the empty set, and
- no self-contact, i.e. a body contacting itself from large deflections.

To ensure that the contacting bodies do not interpenetrate, different contact formulations can be applied. In this manner, *contact compatibility* is enforced by algorithms such as; Pure Penalty, Augmented Lagrange, Normal Lagrange or Multi-Point Constraint. The last two mentioned are only available for asymmetric contact, i.e. only computation of the contact surface pressure and the stress in the *slave* body. Furthermore, the Pure Penalty and Augmented Lagrange formulations utilise more detection point than the others, allowing for a coarser mesh. The drawback of these two formulations is that they allow penetration and therefore a normal contact stiffness must be defined to keep this penetration controlled to some degree. Typically, the Augmented Formulation is favourable due to its accuracy and good convergence. However, the convergence of numerical contact simulations using either Pure Penalty or Augmented Lagrange is heavily affected by the normal contact stiffness factor between two contacting bodies, and this must therefore be tuned to the specific contact problem. The type of contact problem is normally characterised as either; Bonded, Frictionless, Frictional or Rough. These options are used to define tangential sliding and contact stiffness between the bodies.

The problem can be solved as a static structural or explicit dynamics analysis, where the latter requires very small time steps (1e-9) due to the possibility of calculating the influence of an initial velocity before contact-impact, and propagation of stress waves. This analysis is therefore only feasible for a short time period, and a static structural analysis must be applied to calculate the principal stress and contact pressure.

5.2. Relevant advancements in dynamic contact simulation

Simulation of the consequence of normal and tangential velocities of contacting bodies can be used to either simulate microstructural changes in the target geometries (iteratively producing wear) or to estimate local wear coefficients applicable for lumped models. The latter is preferable for the modelling framework, but both will be discussed. Contact fatigue predictions have been designed and applied for, e.g., railroads, gear boxes and ball bearing cages (Van 2013, Ch. 6). Generally, it is a complicated matter to determine local friction coefficients and actual loading conditions, because simple analytical tests cannot be performed. Therefore, controlled test machines and numerical analysis is absolutely necessary (Van 2013, Ch. 6), and yet the predictions includes a high level of uncertainty.

A study of contact and wear with application of Archard's equation is given by (Dhia and Rateau 2004), where the Arlequin framework is used. This allows multiscale and multimodel analysis using a global-local partition of models. Furthermore, it allows the simulation to account for change in the geometry caused by material removal. It is claimed that the Arlequin framework has the highest degree of generality and flexibility compared to other numerical methods. A detailed explanation of how Arlequin overlaps multiple models is given in (Ghanem *et al.* 2013).

A contact-impact algorithm is presented in (Dhia and Zammali 2007), where especially the robustness and stability of the applied contact laws are emphasised. The contact-impact problems are simulated by Lagrangian formulations established from a Level-sets-based Signorini-Moreau model (used to define contact constraints) and the virtual work principle, which is combined with Coulomb's law to include friction. The problem is discretised by time, space and collocation schemes to solve the equations. Numerical examples show stable computation of velocities and material stress and prove the capabilities of a *stabilised* Lagrangian (generalisation of the Lagrangian and Augmented Lagrangian formulations) formulation combined with a level-sets method. This limits the allowed contact penetration and essentially allows for a stable solution verified by numerical examples.

The above methods are computationally difficult and the feasible amount of simulated cycles is limited to around 10^4. These are intended for severe wear situations, which is not the desired mechanism to be analysed. To overcome this limitation, numerical simulation combined with asperity fatigue-based theory have been used to determine the adhesive wear rate of interaction between different materials in (Beheshti and Khonsari 2010). The proposed method utilises the bulk material properties of the softer material along with the friction coefficient and penetration hardness. The simulation model is based on continuum damage mechanical (CDM) theory (reviewed in Section 6.1.4) and outputs a number, N, representing the number of asperity contacts before failure of that given asperity. The application of this method will become clear from Section 6.2.

Among recent advances in discretisation methods and therefore also in simulation of contact problems are the isogeometric analysis (IGA) by Hughes *et al.* (2005). A detailed overview of this method is given in (Cottrell *et al.* 2009). IGA allows integration of CAD and FEA which makes it a strong choice for a modelling framework where different geometries are desired and analysed. Also, if surface roughness is to be included, the smoothness from IGA will become highly relevant as studied by Temizer (2014). Both FEM and IGA are promising discretisation methods, and it is expected that the principles of wear simulation done in FEM problems may be extended to IGA.

6. Modelling of fatigue and wear mechanisms

In part IV and V identification of lumped parameter models are desired, e.g. for the mechanical topology durability. Durability is here a metric for the expected lifetime of the two main mechanical bodies (plunger and seat), which by mechanical stress may experience fatigue. Furthermore, the contact surfaces of the two respective bodies will experience mechanical wear thus degrading the surfaces over time.

Fatigue is in this connection the weakening of materials due to a varying stress and formally defined from the ASTM as: the process of permanent, progressive and localised structural change which occurs to a material point subjected to strains and stresses of variable amplitudes which produces cracks which leads to total failure after a certain number of cycles. The field of material fatigue rates back a few hundred years and it is not the objective to review this historical perspective. However, the milestones regarding design models that are considered in relation to the framework are outlined in Section 6.1.

Wear is on the other hand defined as: 'the progressive loss of material from the operating surface body occurring as a result of relative motion at its surface' by the OECD (Lansdown and Price 1986) and may also come into play for ACVs. This is addressed in Section 6.2. As described in Section 3, the classical design approach is to consider material fatigue, where different approaches may be used.

6.1. Fatigue models

The most common approach is a strain-based (Wöhler/*S-N*) analysis combined with Goodman diagrams in case of varying load profiles. This approach has been applied throughout history in many different mechanical structures, e.g. (Ritchie and Lubock 1986, Marczewska *et al.* 2006, Malakizadi *et al.* 2007, Zhao *et al.* 2017).

Studies of Very High Cycle Fatigue (VHCF) of metals (Bathias 1999, Murakami *et al.* 1999, Nishijima and Kanazawa 1999), have shown that many metallic materials have no infinite fatigue life. It is claimed that most modern machines have their fatigue limit in the range of 10^9–10^{10} cycles, which is in the range of what the ACVs are expected to withstand. This is of course influenced by the type of loading, and the majority of VHCF tests are done with fully reversed loads. The resulting fatigue endurance limit for this amount of loadings is within 300–500 MPa which may serve as a constraint when optimising the ACV. The different stages of the fatigue process in metallic materials are shown in Figure 4. Depending on the region (I, II or III), different damage laws are used (Lemaitre 1984).

A comprehensive study of cumulative fatigue models is given in (Fatemi and Yang 1998) where the following conclusion is given: *Though many damage models have been developed, unfortunately, none of them enjoys universal acceptance. Each damage model can only account for one or several phenomenological factors, such as load dependence, multiple damage stages, non-linear damage evolution, load sequence and interaction effects, overload effects, small amplitude cycles below fatigue limit and mean stress.* However, a number of models are highlighted, i.e. the phenomenological-based linear and non-linear damage rules and the analytical crack growth concept approaches. Some are described below.

6.1.1. The Palmgren-Miner law

In the area of mechanical design the Palmgren-Miner law (*linear rule*) (Miner 1945) shown in Equation (6) and variations hereof have been widely used for predicting fatigue of cyclic-loaded mechanical components.

$$C = \sum_{i=1}^{k} \frac{n_i}{N_i} \tag{6}$$

where n_i is the accumulated number of cycles at a repeated stress pattern (s_i), N_i is the average number of cycles until failure at s_i and C is the damage fraction indicating failure upon reaching a value of 1. The total damage is a sum of damage caused by possibly varying

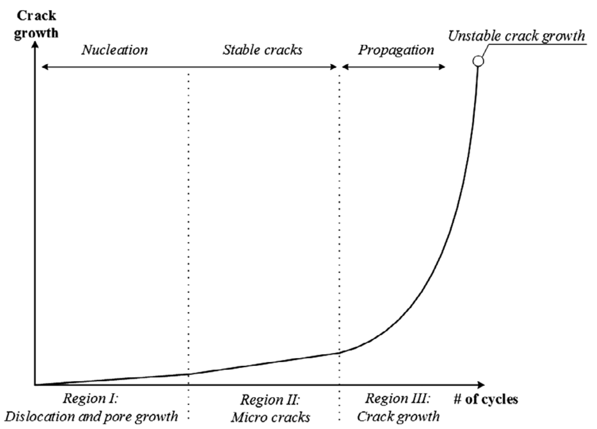

Figure 4. 'Fatigue life conventional stages in the metallic materials' with inspiration from Oller *et al.* (2004).

load cycles, and by introducing cycle counting methods of *RainFlow*-type complex loading conditions can be included in the fatigue prediction (Chaboche 2013, Ch. 3). This accumulated fatigue prediction is based on experimental test data performed under different loading conditions.

6.1.2. The Manson-Coffin law

Generally, the Manson-Coffin law shown in Equation (7) (Manson 1954, Coffin 1960) is applied to predict fatigue life in the low cycle fatigue (LCF) regime, which in general is below 10^5 cycles (depends on material). A more accurate distinction may be established by examining the material cracks. In a detailed study (Brechet *et al.* 1992), it is observed that surface nucleation and density of micro-cracks is dominant in LCF, nucleating after 10% of the LCF life compared to 60% for HCF. This may be predicted by:

$$N_{f,p} = \frac{1}{2}\left(\frac{\Delta\epsilon_p}{2\epsilon'_f}\right)^{-c} \tag{7}$$

where $N_{f,p}$ is the predicted number of cycles before failure, $\frac{\Delta\epsilon_p}{2}$ is the plastic strain amplitude, ϵ'_f is the fatigue ductility coefficient and c is the fatigue ductility exponent.

Wang *et al.* (2017) compares the use of the S–N curve with the Manson-Coffin law in the VHCF regime above 10^7 cycles to predict the fatigue process. Ultrasonic testing equipment is used to reach the required cycles and using full-field temperature measurement by infrared thermography, the fatigue process is monitored in terms of specimen temperatures. Results show that the Manson-Coffin law outperforms prediction of fatigue by use of the S–N curve. Thus, making it an interesting model to predict fatigue of the ACV.

6.1.3. Fracture mechanics

Griffith (1921) contributed to the field of Fracture Mechanics (FM) with the idea of using energy equilibrium to describe the initiation and growth of cracks. This later evolved to the definition of the J-integral by Cherepanov (1967) and Rice (1968) independently, showing that a line path integral was equal at any path around the tip of a notch in a 2D strain-field. The integral represents the energy release rate for crack growth and has proven to be a strong formulation for use in FEM.

6.1.4. Continuum damage mechanics

Kachanov (1958) and Rabotnov (1968) introduced the concept of CDM, which was applied by Chaboche (1980) and Lemaitre and Chaboche (1994) to formulate an equation for nonlinear damage evolution, serving as the foundation for the state-of-the-art CDM models, e.g. (Alfredsson and Stigh 2004, Oller *et al.* 2004, Darabi *et al.* 2011). For the sake of brevity, the damage evolution laws are left out. Three main advantages of CDM are that it accounts for growth of damage below the fatigue limit, initial hardening effect and incorporation of the mean stress. CDM models are essentially based on a thermodynamic framework along with a principle of strain equivalence. A damage variable which decreases from 1 (for a virgin material) towards 0 determines failure. In a CDM framework, failure does not mean fracture. Rather, it indicates a point at which the material has adequately degraded as a result of micro-cracks and micro-voids formation (discontinuities) (Beheshti and Khonsari 2010). The method has recently found use for material degradation and fatigue simulation of contacting asperities to determine the amount of cycles until a wear particle is produced (Bryant *et al.* 2008, Beheshti and Khonsari 2013).

6.1.5. Hertzian contact stress

The practical application of the above models requires knowledge about material stresses. In contact mechanics, this type of stress is a complex matter to compute, and a simple formulation with satisfying results is sought. Hertz theory and modifications hereof is a well-proven approach, and still part of the state-of-the-art in spite of the inherit limitations and is therefore an interesting approach. One shortcoming of this method is that it is limited to relatively simple and idealised geometries. However, the valve contact problem is similar to that of two tori (with minor radii around the x-axis and major radii around the z-axis, r_1, r_2, R_1, R_2). This geometry is further simplified to the contact of two cylinders with length $L = 2\pi R_1 = 2\pi R_2$, which may aid in revealing a simplified relationship between the maximum pressure and valve geometry. The pressure distribution on the contact surfaces was proposed by Hertz as:

$$p(x) = p_0\sqrt{\left(1 - \frac{x^2}{a^2}\right)} \tag{8}$$

where a is the contact half-width which for a cylinder on cylinder contact is by Equation (9), σ_0 is the maximum pressure given by Equation (10) and x is the contact length from the centre of contact.

$$a(F_n) = \left(\frac{2F_n r^*}{\pi^2 RE^*}\right)^{1/2}; \frac{1}{E^*} = \frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}; \quad \frac{1}{r^*} = \frac{1}{r_1} + \frac{1}{r_2} \tag{9}$$

$$p_0(F_n) = \frac{F_n}{\pi^2 Ra(F_n)} = \left(\frac{F_n E^*}{\pi^2 Rr^*}\right)^{1/2} \tag{10}$$

where E^* is a factor including Young's modulus of elasticity (E) and Poisson's ratio (ν) for the respective materials, r^* is a factor accounting for the radii of contacting tori and F_n is the normal force. The underlying assumptions of the Hertz theory are (Johnson 1985):

- The surfaces are continuous and non-conforming and the strains are small: $a << r^*$;
- Each solid is considered an elastic half-space; and
- Both surfaces are frictionless.

The maximum pressure from Equation (10) reveals an inverse quadratic relationship between the normal load and the maximum pressure depending on the major and minor radius of the contacting tori, thus the contact area plays a significant part in reducing peak material stresses. Hertz theory does not allow tensile force when separating two elastically deformed solids, in reality adhesion will occur even in the absence of surface roughness. The contact stress from Hertz theory is not sufficient to describe these adhesive forces, but with relatively simple modifications adhesion can be included (Johnson 1985).

In summary, the appropriate fatigue model depends on the application and theirs prediction will come with a certain range of uncertainty. The proposed methodology for ACVs is to consult some of the presented methods and compare the predicted cycles until failure. This range of cycles until failure may serve as a design metric.

6.2. Wear mechanism modelling

A detailed explanation of wear modelling and application of wear maps is given by Williams (1999, 2005). He underlines that: 'No simple and universal model is applicable to all situations'. Furthermore, almost 200 'wear equations' have been proposed, yet none predicts the tribological performance with certainty and confidence. Couplings between different types of wear, and sudden transition zones from mild to severe wear will be present when the entropy of a tribological system increases. Therefore, there exists a gap of accelerated lifetime test techniques, which may be systematically applied to such system (Bryant *et al.* 2008).

One wear equation is highlighted, namely Archard's wear equation, which has found widespread use. This was originally intended to predict adhesive wear, but also abrasive wear has shown to be captured by this model. Details about this model are given in Section 6.2.1. In order to elaborate on the wear occurring in ACVs, a recap on mechanical wear processes is presented. Wear is divided into four categories as shown in Figure 5.

Definition of the four types reads (Straffelini 2015, p. 100):

- Abrasive wear: describing the interaction (two or three bodies) of a hard particle/protuberance plastically penetrating a softer counterface and groove it.
- Erosive wear: describing an extremely short sliding executed within a short-time interval, e.g. particles in a fluid impacting a surface (with velocities in the m/s range).
- Adhesive wear: describing the mechanism when adhesion forces between contacting asperities exert a predominant role in formation of wear fragments.
- Surface fatigue: describing the mechanism of cyclic loading where a crack is nucleated and later propagated to a final fracture.

The ACVs are subject to a mechanical wear process with a repeated load at high frequencies (with a rated motor/pump speed of 800 rpm each valve has a cycle every 75 ms with potentially large impact forces and pressure differentials of 345 bar). The system comprises a hydraulic fluid, which will introduce a hydrodynamic dampening at end positions and also making it a mixed-friction contact problem. Furthermore, the oil flow and additives of the oil will influence the mechanical components. Definitions of the different severities are detailed in (Williams 2005) and these are used together with engineering intuition to reveal the assumed wear phenomena after break-in in prioritised order as:

- *Pitting*: Impact of plunger and seat causes subsurface plastic shear, thus propagating cracks nucleating from pre-existing voids or inclusions present in the material structure.
- *Ratcheting*: Cycle-by-cycle local incremental plastic strains can produce local surface fatigue due to cracks parallel to the surface, with lamellar or sheet-like formed debris particles (less than a micron thickness) (Williams 2005).

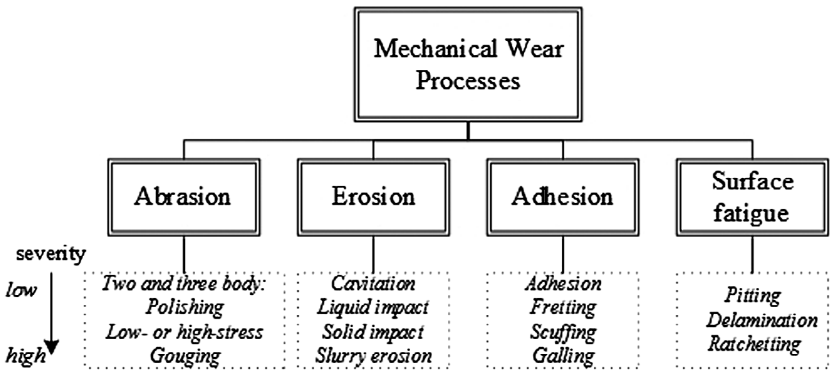

Figure 5. Mechanical wear mechanism diagram. Picture with inspiration from Williams (2005).

- *Fretting*: Adhesive wear from micro-sliding in the VSI.
- *Polishing*: Abrasive wear during the break-in period decreasing the surface roughness.
- Mixed *liquid and solid impact*: Erosive wear related to the flow through the valve seat (severity depends on fluid contamination).

The most obvious fault mode, which is a consequence of these wear mechanisms is leakage at the seat resulting in lower DDM efficiencies. More severely, failures may occur upon accumulation of wear particles leading to fluid contamination, which potentially may cause system breakdown by either accelerating the mechanical wear or by interfering with the moving coil actuator in worst case disabling the capability of actively closing the ACV. Surface fatigue from repeated loading, material imperfections or misalignment of the armature may cause an abrupt fracture resulting in an uncontrollable system. This failure is highly influenced by the micro-geometry.

Mechanical wear prediction of seat valves used in combustion engines (both inlet and exhaust), have gained increasing interest in the past decade (Lewis *et al.* 1999, Lewis and Dwyer-Joyce 2001, 2002, Chun *et al.* 2006, Cavalieri *et al.* 2014, 2015, Oh *et al.* 2014). These valves comprise similar mechanisms as ACVs, i.e. impact wear and seat compression. Lewis *et al.* (1999) presents a study of wear mechanisms in inlet valves revealing that impact of the plunger on the seat during closure and sliding between the two surfaces are the main wear mechanisms, i.e. abrasive and adhesive wear mechanisms. Studies (Lewis and Dwyer-Joyce 2001, 2002) further show that for this specific type of valves impact wear is approximately proportional to the square of impact velocity. The tests are conducted with and without lubrication and the wear recession is observed to be approximately 3.5 times larger under dry conditions, and the wear with lubrication is stated to be 'barely visible'. Oh *et al.* (2014) states that the most common failure of seat valves used in combustion engines are caused when contact between armature and seat becomes uneven. During the closing event, partial contact and combustion pressure result in high stress concentrations in the valve neck, causing a fracture. The issue originates from the high engine shaft speed meaning high impact velocities occurring at high temperatures (400 °C) deteriorate the material strength. This mechanism is not expected to occur in the typical ACVs due to obvious design differences.

The most recent application of Archard's equation in relation to valves includes Cavalieri and Cardona (2012) and Cavalieri *et al.* (2015) whom determines wear in combustion seat type valves using Archard's law in combination with a mortar contact algorithm.

6.2.1. Archard's equation

Archard (1953) established a definition of unlubricated wear related to the dissipated frictional work:

$$W_s = sK\frac{F_n}{H} \tag{11}$$

where W_s is the volume of removed material due to sliding, s is the sliding distance, K is a dimensionless coefficient based on the probability of production of wear per unit encounter, F_n is the normal force and H is the Vickers indentation hardness of the softer material. Equation (11) yields that the wear rate is independent of the apparent area of contact and that the wear rate is directly proportional to the load as stated in (Archard and Hirst 1956). However, this was a misleading conclusion by Archard and Hirst (1956), since increased normal load, sliding speed or bulk temperature leads to a sudden transition phase from mild to severe wear (Williams 1999). This non-linear behaviour and the transition from mild to severe wear have been undefined until a recent study of adhesive wear done by Aghababaei *et al.* (2016). It is shown that there exists a transition in the asperity wear mechanism when contact junctions are below a critical length scale. This determines whether a gradual smoothing of the surface or a fracture-induced debris, evolves from the adhesion, and this is predicted by an analytical expression. This is a step towards eliminating the requirement of empirical wear coefficients for adhesive wear modelling. Another major achievement towards eliminating empirical wear coefficients in mixed lubrication wear modelling is presented by Beheshti and Khonsari (2013). The presented approach predicts wear by a modified version of Archard's law that uses the probability of one asperity contacting another asperity. Furthermore, the model uses load sharing between fluid and asperity, where the load carried by the fluid does not contribute to any wear. The wear coefficients are predicted by the numerical methodology in (Beheshti and Khonsari 2010), which essentially utilises CDM and thus considered the adhesive wear as a fatigue phenomenon.

Adhesive wear depends on the sliding distance and the normal force at which this sliding occurs. Here, the materials in contact and the surface finish play a crucial role in the resulting wear. The current ACV topology is designed without any flow surfaces intended to undergo sliding. However, it is relevant to study the design sensitivity if a certain amount of sliding does occur and how asperities interact during the ACV closing event.

6.2.2. Wear caused by impact

During the closing event of ACVs, high impact velocities are expected due to the low switching time. The wear from this phenomenon, as included in part IV from Figure 2, is here elaborated.

Lewis and Dwyer-Joyce (2002) applies Equation (12) to determine the impact wear, W_i, in an internal combustion engine valve with inspiration from Fricke and Allen (1993) who again based the impact wear model on the model presented by Hutchings *et al.* (1976). In

(Hutchings *et al*. 1976), erosion wear of metals caused by solid particles, was described as:

$$W_i = K_i N e^n \tag{12}$$

where K and n are empirically determined wear coefficients, N is the number of cycles and e is the kinetic impact energy:

$$e = \frac{1}{2} m v^2 \tag{13}$$

where m is the mass of the moving member and v is the impact velocity. Interpreting this, yields that the impact velocity has a quadratic contribution which is further raised to the power of n. This exponent may be chosen below one, resulting in a scaled contribution from the kinetic energy according to the application. This emphasises the importance of knowledge about impact wear, impact velocity and possible reduction of this. The above presented model has been tested at impact velocities between 0.324 and 3.68 m/s with a significant recession at velocities above 1 m/s by Lewis *et al*. (1999). On the other hand, Clark *et al*. (2005) states that the limit in impact velocity is around 0.05 m/s for the same application. This velocity is depending upon the required number of repetitions and the test environment, which is why these limits cannot necessarily be trusted as design parameters.

6.2.3. Erosion wear modelling

If external environmental factors such as fluid contamination particles are present in an ACV, the flow edges may erode and wash out over time leading to internal leakage flow in the valve as discussed by Zhang *et al*. (2013). An early contribution to the field of erosive wear is presented by Hutchings *et al*. (1976), which also applied to impact wear as shown above. Common practice is to describe the wear as the ratio between the mass of removed material and the mass of impinging particles (Njobuenwu and Fairweather 2012). ANSYS highlights the model originating from Edwards (2000) with the following formulation for the erosion rate of wall boundaries:

$$ER = \sum_{p=1}^{N_p} \frac{\dot{m}_p C(d_p) f(\alpha_p) v_p^{b(v_p)}}{A_{\text{face}}} \tag{14}$$

where A_{face} is the face area at which N_p number of particles impact, $C(d_p)$ is a function of particle diameter, v_p is the relative particle impact velocity, $b(v_p)$ is a function of particle velocity, $\dot{m}_p$ is the mass flow rate of the particle and $f(\alpha_p)$ describes the dependence of erosion on the particle impact angle, α_p. This expression is currently used for erosion modelling by ANSYS Fluent (Agrawal and ANSYS 2012, p. 44), who states that a universal model is not realistic but that multiphase CFD may be utilised to gain insight for erosion predictions.

The review of erosion presented in (Humphrey 1990) emphasises that the wear is influenced by the flow profile (laminar/turbulent), particle's momentum equilibrium number and temperature. The phenomenon is argued to be a complex matter to model, and around 28 models for solid particle-wall erosion have been proposed. None of the attempts have a universal applicability and thus relies on experimental data (Njobuenwu and Fairweather 2012).

Recent advances in the field include Zhang *et al*. (2013), who presents a lifetime prediction of an electro-hydraulic servo valve. The main focus is to simulate the wear rates caused by erosions. The approach is based on CFD and erosion theory, where Equation (14) is applied. In general, the results reveal a feasibility of the proposed method in assessing the degradation trend of the valve performance. For an ACV, it would thus be necessary to estimate the amount of fluid contamination particles and the velocity at which these impact the flow edges. There is no obvious method to obtain this information. Another example of erosion occurring in choke valves is given by Noegkleberg and Sontvedt (1995). In here, it is stated that loss of material is observed around the vena contracta, and the cause is presumed to be cavitation. The geometrical difference from using an annular seat valve is expected to decrease the possibility of severe fluid erosion wear.

In summary, it is clear that prediction of the dominating wear mechanisms is not trivial (due to; macro- and micro-geometry, loading conditions, materials, coatings, manufacturing and environmental affections), in fact several authors claimed that design of arbitrary systems based on universal wear models was far in the future (Hsu *et al*. 1997, Williams 1999). Since then, one of the most groundbreaking ideas is to relate the entropy change to the irreversible degradation process (Bryant *et al*. 2008, Beheshti and Khonsari 2013), which has provided theoretical techniques to estimate adhesive wear. Another recent study of the interaction between asperities (Aghababaei *et al*. 2016) showed that transition from mild to severe wear occurs when the junction size reach a critical value. These methods still have not found widespread use.

A literature review of wear mechanisms in seat valves used in combustion engines revealed a consensus on application of Archard's wear law; in particular, modelling of unlubricated micro-sliding of two surfaces. Furthermore, wear associated with impact of two bodies is strongly correlated to the kinetic energy at impact, although geometry, material combination and impact angle are influencing this mechanism. This wear associated with impact is described under dry conditions, and further work must be devoted to investigate the squeeze film effect on the metal on metal contact, but equally important the effect of the compression forces related with the presumed thin film fluid with locally high pressures. Depending on fluid contamination and wear, a wash out of fluid edges may be experienced over the lifetime of the ACV thus introducing an internal leakage that is consequently affecting the efficiency of DDMs and challenging the feasibility of hydrostatic transmission.

7. Conclusions

A conceptual modelling framework of ACVs has been presented in Figure 2, which fits into the methodology presented in Figure 1. The framework was divided into five parts and reviews of relevant methods have been presented. Based on the reviews, the following conclusions may be established with regard to possibilities and limitations toward a modelling framework for ACVs.

In regards to simulation of fluid dynamics, both numerical and analytical techniques are necessary. It is expected that a CFD model can be used to evaluate flow characteristics and fluid forces on the plunger. The computational effort connected to this is an issue, and establishment of transient models is time-consuming. Therefore, analytic alternatives for determination of flows and movement-induced forces are of high interest and a comparative study of this is required. Lumped parameter and analytic models of movement-induced forces, flow forces and squeeze film damping and stiction forces have been identified. Modelling of thin film fluid has a drawback that small gap heights requires increasingly smaller time steps and may therefore become computationally inefficient. The minimum necessary height is not easily determined, but this plays a role in understanding the material fatigue during contact-impact of the VSI, and the collapse of the fluid must be further studied.

Material stresses including contact-impact, squeezing and adhesion in the valve seat insert have to be obtained either by numerical analysis or from a Hertz framework in order to facilitate the features required in the design framework. Factors such as residual stresses and tensile loads are of interest in this regard, and computational feasibility is prioritised compared to accuracy.

Several uncertainties regarding wear modelling and gaps in the knowledge about wear and fatigue phenomena in general, raises a concern about the feasibility of predicting lifetime of ACVs. The combination of contact mechanics and fatigue in the gigacycle regime makes such a prediction unreliable with . However, based on findings from (Lewis *et al*. 1999, Lewis and Dwyer-Joyce 2001, 2002, Oh *et al*. 2014, Cavalieri *et al*. 2015) contact-impact and contact pressures will be significant and modelling techniques for approximation of this exist, mainly studied under dry conditions. Adhesive wear, as seen in combustion engine valves (Cavalieri *et al*. 2014, 2015) will be present in ACVs if tangential slip is allowed in the VSI, e.g. from inclined flow edges. Therefore, a sensitivity study of tangential slip at the surfaces and impact velocities is considered useful to give a relative measure of expected wear, where the approach presented in (Beheshti and Khonsari 2013) has the desired capabilities required in mixed-lubrication wear.

Studies of spool-type servo valve have shown erosion to be an important mechanism when predicting lifetime of such. This type of wear does depend on fluid contamination, particle velocity and areas of affected flow edges. The level of contamination is not known, but if contact wear is significant there will be an increasing amount of debris in the fluid over time. The flow in an annular valve is characterised by large opening areas, leading to low fluid velocities and a laminar flow in most of the operating time. Thus, erosion from debris will be limited by these large opening areas, it is though necessary to study this further at the current state of research.

The conclusions regarding wear mechanisms are not backed up by data for this specific valve, since such data is not available yet. Furthermore, the availability of a systematic test procedure to accelerate tests of the current valve prototype is non-existing and extensive work is required to achieve such.

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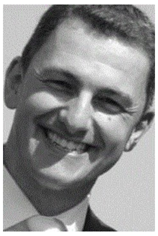

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