

Physically Motivated Model for Efficient Dynamic Simulation of Chain Tensioners with Labyrinth Seals

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ABSTRACT

The object of this study is a new chain tensioner with two labyrinth seals. For the simulation of chain tensioners within the framework of multi-body dynamics, a physically orientated model to describe the fluid dynamics of the labyrinth seals is derived. The easiest way to describe labyrinth seals is to use maps obtained from measurements. As this is very time-consuming, methods of 1D and 2D fluid-mechanics are used in this work to model the labyrinth seals. The seals are characterized by physically motivated parameters e.g. coefficients of resistance or friction. As these parameters can be derived from geometric data, a very good forecast feasibility without experimental investigations is provided. For high accuracy simulations model parameters can be refined by experimental data. As many and highly complex parameters have to be identified, this refinement is very time-consuming and requires lots of experiments. Therefore, a third approach for modeling a labyrinth sealing is derived. Using dimensional analysis the labyrinth can be described by a non-dimensional equation. Only a few coefficients have to be determined by measurements. Hence, the effort for parameter identification as well as the number of necessary experiments is significantly reduced. All three approaches are validated with experimental data. Next, a dynamic model of a complete chain tensioner including labyrinth seals was built up. A comparison between simulation and measured data of the flow characteristics as well as of the dynamic behavior is presented to prove accuracy, benefits and practicability of the presented approach. Furthermore, the influence of fluid inertia in the hydraulic lines is analyzed.

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INTRODUCTION

Valve trains are used in combustion engines to control the combustion process. To ensure a synchronous movement of valves and pistons, the crankshaft is connected to the camshaft within the timing assembly. Chain drives are well suited for the transmission of high power and have no need for maintenance. Therefore, modern combustion engines make more and more use of chains as timing elements yielding complex oscillation systems. Vibrations are excited in many ways, e.g. by alternating torques from gas and combustion forces, valve actuation, inertia forces, eigenmodes of shafts and guide elements or the polygon effect caused by chain and sprocket. The hydraulic chain tensioner is of great importance for the reliable operation of a timing chain drive. It has to provide the necessary pretension on the chain drive and compensate thermal elongation and wear to avoid timing deflections or even a jump over at the sprocket. To ensure vibration-free and quiet operation, the tension system must also provide appropriate damping to de-energize resonance phenomena or forced vibrations and stabilize chain oscillations with minimal friction. The required damping is generated by leakage mechanisms of the hydraulic system. To design specific damping characteristics and frequency dependencies, sophisticated approaches use labyrinth seals which are the focus of this work.

As the tensioner force is directly linked to the chain force, the tensioner is the most important component to influence the dynamic behavior of the timing drive and to keep chain forces in the desired range with respect to friction, acoustics and life time. For optimization and target orientated design detailed simulation models of the chain tensioner covering different physical domains, i.e. mechanics and hydraulics, are needed.

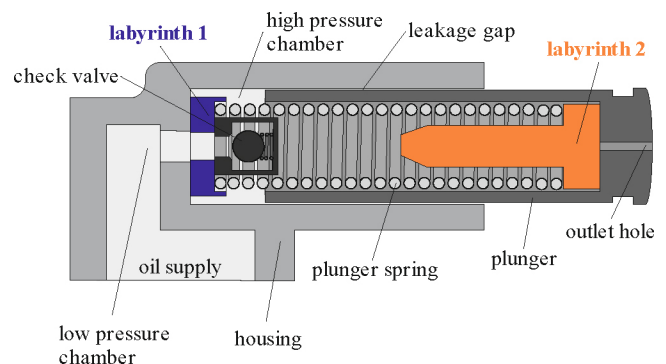


Figure 1. Hydraulic chain tensioner with two labyrinth seals.

Object of this study is a new chain tensioner with two labyrinth seals (cp. Fig. 2). In previous works [1,2], the dynamical flow behavior through the labyrinth seals has to be determined by means of 1D simulations or time-consuming and laborious experimental

measurements. Hence, the aim of this work has been to derive a physically orientated model for labyrinth seals that is usable for dynamic simulation of chain tensioners within the framework of multi-body dynamics and hydraulics and describes the frequency dependent behavior.

In the first part of this article a new physically orientated model comprising three approaches for different modeling depths is presented and compared to measurements. Instead of time-consuming CFD simulations or measurements, methods of 1D and 2D fluid-mechanics are used. The seals are characterized by physically motivated parameters e.g. coefficients of resistance or friction which can be derived from geometric data providing a very good forecast feasibility. For high accuracy simulations the model parameters are refined by experimental data. As many and highly complex parameters have to be identified, a third approach based on dimensional analysis is derived that is best suited for automatic curve fitting instead of complex parameter identification. This approach only requires a minimum number of measuring points. The validity of all three approaches is proven with experimental data.

The derived models of the labyrinth seals are used to model a state of the art chain tensioner. A comparison between simulation and measured data of the dynamic and frequency dependent behavior of the chain tensioner is presented to prove accuracy, benefits and practicability of the presented simulation model. For this, investigations were carried out in a wide temperature and frequency range. To collect the experimental data, two different hydraulic test-rigs were used: one for measuring the static flow characteristics of the labyrinth seals and one for dynamic measurements of the chain tensioner (excitation in time domain). Both test rigs are presented in this work.

DESCRIPTION OF CHAIN TENSIONER

The functional principle of a hydraulic chain tensioner is to combine a spring loaded plunger with a hydraulic system that allows to increase and adjust the preload force as well as the stiffness characteristics. Furthermore, the hydraulic system creates the desired damping with defined leakage mechanisms which are the primary cause of the frequency dependent behavior of the chain tensioner. The easiest way to implement damping is a leakage gap between housing and plunger. To design specific damping characteristics sophisticated approaches use additional labyrinth seals.

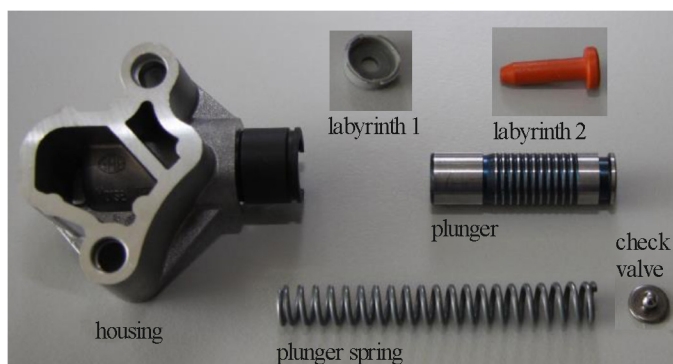


Figure 2. Hydraulic chain tensioner: disassembled into components.

The schematic structure and components of the investigated chain tensioner are illustrated in Fig. 1 to 3. The basic composition is similar to standard chain tensioner. A plunger is guided in a housing with a plunger spring mounted between. A ball check valve separates the high pressure chamber (HPC) and the low pressure or supply chamber (LPC). Engine oil supplied to the low pressure chamber passes through the check valve in the high pressure chamber. If the tensioner is squeezed, the valve closes and the oil is pressed out through a leakage gap. The most common design is only one straight leakage gap between piston and housing. The investigated tensioner has two additional labyrinth seals: the first as parallel circuit to the check valve (labyrinth 1) and the second as parallel circuit to the conventional leakage gap (labyrinth 2). Labyrinth 2 is placed at the top of a plastic element that also serves as packing material to adjust the volume and stiffness of the high pressure chamber.



Figure 3. Investigated labyrinth seals.

Two phases can be distinguished during operation: compression and expansion. During the compression phase the chain forces lead to a contraction of the tensioner. The contact forces acting on the tension blade push the plunger against the stiffness of the spring and hydraulic volume. The check valve closes and the pressure in the HPC rises. The enclosed oil is pressed through the labyrinths and leakage gap. In the expansion phase the plunger extends and the pressure in the HPC decreases. The check valve opens and the oil loss is compensated by the supply line.

MATHEMATICAL MODEL

To describe the dynamical behavior of modern timing assemblies, different physical domains - e.g. mechanics, hydraulics and optionally control - must be taken into account and integrated in one consistent mathematical model. The theoretical framework and governing equations are presented in this section.

Multi-Body System Dynamics

The system dynamics are based on the equations of motion for multibody systems extended by hydraulic equations. The basic equations of motion for general multi-body system consisting of mass carrying bodies and massless force elements write:

$$\mathbf{M} \cdot \ddot{\mathbf{q}} = \mathbf{h}.$$

(1)

The symmetric and positive definite mass matrix $\mathbf{M} = \mathbf{M}(\mathbf{q})$ depends on the generalized positions $\mathbf{q}$. The vector $\mathbf{h} = \mathbf{h}(\dot{\mathbf{q}}, \mathbf{q}, t)$ denotes all external, internal and gyroscopic forces depending on $\mathbf{q}$, the generalized velocities $\dot{\mathbf{q}}$ and time t . A general mathematical description of multi-body systems can be found in [3].

Hydraulic Systems

Hydraulic systems are considered to be a network of basic components such as valves, hydraulic lines and hydraulic cylinders. Abstracted to a logical structure, hydraulic systems comprise nodes and links. The behavior of nodes is described by the pressure p as a possible state variable and volume V . Links that connect hydraulic volumes or mechanical systems are characterized by their volume flow Q as a possible state variable. Considering compressibility of the fluid, mass conservation results in a non-linear differential equation for nodes:

$$\dot{p} = \frac{K}{V} (\sum_j Q_j + Q_{\text{mec}}) \quad (2)$$

$$\text{with } K = -V \frac{\partial p}{\partial v}. \quad (3)$$

Therein Q_j denotes the hydraulic volume flow into the node and $Q_{\text{mec}} = \mathbf{W}_{A,Q} \cdot \dot{\mathbf{q}}$ the variation of the volume due to mechanical systems such as hydraulic pistons. $\mathbf{W}_{A,Q}$ represents the area of a mechanical interface to the hydraulic volume projected in the direction of the generalized coordinates of a connected multi-body system. The bulk modulus K depending on the air content and the pressure describes the compressibility of the fluid. Hydraulic links with cross section area A and length l relate the change of flow $\dot{Q}$ with the connected pressures (p_{in} , p_{out} and pressure loss Δp) using momentum conservation:

$$\dot{Q} = \frac{1}{\rho l} A (p_{\text{in}} - p_{\text{out}} - \Delta p) \quad (4)$$

$$\text{with } \Delta p^k \propto Q. \quad (5)$$

The exponent k relates pressure loss and flow. It is equal to 1 for laminar flow (Hagen-Poiseuille) and 0.5 for energy conservation according to Bernoulli's principle. Neglecting the inertia of the moving fluid mass, the flow Q is given by the algebraic Equation (5). The reaction forces generated by the fluid pressure p to the mechanical system are given by the generalized hydraulic force $\mathbf{h}_{\text{hmi}}$:

$$\mathbf{h}_{\text{hmi}} = \mathbf{W}_{A,Q}^T \cdot p. \quad (6)$$

Combining the mechanical and hydraulic equations, the differential equations for multi-body systems including first order systems are written as:

$$\begin{pmatrix} \mathbf{M} & 0 & 0 \\ 0 & \mathbf{M}_Q & 0 \\ 0 & 0 & \mathbf{M}_p \end{pmatrix} \begin{pmatrix} \ddot{\mathbf{q}} \\ \dot{p} \\ \dot{Q} \end{pmatrix} = \begin{pmatrix} \mathbf{h} + \mathbf{h}_{\text{hmi}} \\ \sum_j Q_j + Q_{\text{mec}} \\ p_{\text{in}} - p_{\text{out}} - \Delta p \end{pmatrix} \quad (7)$$

$$\text{with } M_{Q,ii} = \frac{V_i}{K_i}, \quad M_{p,ii} = \frac{\rho l_i}{A_i}. \quad (8)$$

As seen in Equation (7), the mathematical structure of the equations of multi-body systems with and without hydraulic components is equal and therefore the same integration schemes can be used.

One main problem of the simulation is that for small volumes and nearly incompressible fluids such as oil this approach leads to very stiff differential equations. Hence, the differential equation (2) can be substituted by an algebraic equation assuming an incompressible hydraulic volume. For an incompressible node only the mass conservation equation has to be satisfied as a constraint:

$$\sum_j Q_j + Q_{\text{mec}} = 0. \quad (9)$$

LABYRINTH SEAL

This chapter addresses the description of the flow conditions within the labyrinth seals in terms of pressure losses. To avoid extensive CFD simulations a physically motivated model for the labyrinth seals is introduced. Therefore, the fluid flow within the labyrinth seals is analyzed and an analytical description for the flow rate as a function of the pressure loss is derived. Three approaches with different modeling depth, range of application as well as necessary time and effort for data collection are presented:

- Predictive physical model
- Calibrated physical model
- Non-dimensionalized model

Test Rig for Labyrinth Seals

To get experimental data a specific test rig was built (Fig. 5). The labyrinth seal is placed into a special designed adapter with installation situation comparable to the chain tensioner. Conditioned oil is supplied to the labyrinth seal through the adapter. Within the adapter, a pressure and a temperature sensor are placed in front of the labyrinth seal. An additional temperature sensor is installed at the oil outlet. The mass flow rate is continuously measured with an automatic weighing scale below a collecting tank. The oil pressure is generated by a hydraulic pump driven by a servomotor. With the help of a control valve in combination with the servomotor the pressure is controlled to a constant value up to 90 bar. After preheating the oil in the storage tank, the exact temperature is controlled by an electric cartridge heater in front of the adapter in the range from room temperature up to 110°C. For constant oil properties an ultrasonic based degassing unit is integrated in the storage tank. Measuring is done after steady-state conditions are reached.

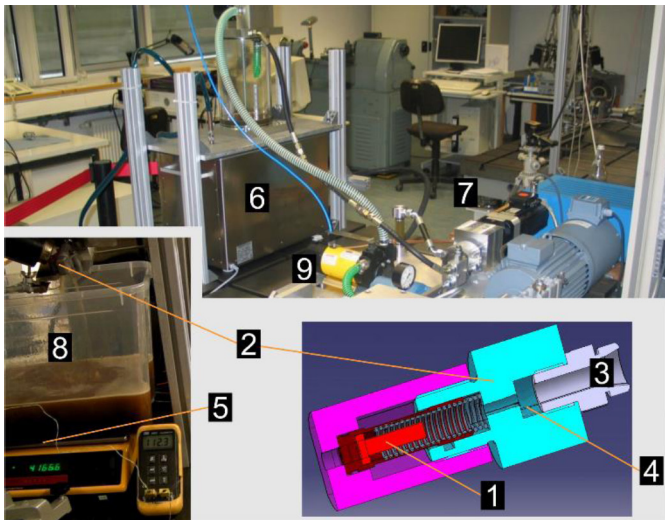


Figure 5. Test rig for labyrinth seals (1 labyrinth seal, 2 adapter, 3 hydraulic connection, 4 measuring point pressure and temperature, 5 scale for mass flow measurement, 6 oil storage tank and conditioning unit, 7 hydraulic pump with servo drive, 8 collecting tank, 9 scavenger pump).

Simulation and Experiment: Labyrinth Seals

The easiest way to describe the fluid dynamics of a labyrinth seal is to use a map. As a lot of measurements at different conditions (temperature, pressure, oil viscosity, ...) are necessary, the generation of such a map is very time-consuming and labor intensive. To avoid this, a different approach is chosen in this work: The labyrinth seals are characterized by only a few physically motivated parameters e.g. coefficients of resistance or friction. These parameters can be derived from geometric data providing a very good forecast feasibility without experimental investigations. Thus, layout options, parameter studies and system optimizations can be performed very easily in an early stage of the design process. This approach is called **predictive physical model** within this work. For detailed high accuracy simulations of existing labyrinth seals a second approach is used: The model is refined by experimental data and model parameters are adjusted to the measurements. This approach is referred to as **calibrated physical model**. As many and sometimes highly complex parameters have to be identified, this refinement is very time-consuming and requires lots of experiments. Therefore, a third approach for modeling a labyrinth sealing is derived: **non-dimensionalized model**. Using dimensional analysis the labyrinth can be described by a non-dimensional equation. All pressure losses are summarized in one functional expression which depends only on the Reynolds number. Only a few coefficients have to be determined by a curve-fitting algorithm. Hence, the effort for parameter identification as well as the number of necessary experiments is reduced significantly. All three approaches are presented in detail in the next sections and validated with experimental data of two different labyrinth seals.

Approach 1: Predictive Physical Model

The idea behind this approach is to reduce the labyrinth seal to a hydraulic conduit system and analyze it as hydraulic circuit. The total hydraulic resistance is composed of single elements based on 1D and 2D fluid mechanics. The relevant resistance coefficients can be taken from tables and reference books: [6], [13] and [15]. The hydraulic resistance and corresponding pressure losses Δp in the course of motion of the fluid are mainly caused by two types of fluid losses: frictional losses ($\Delta p_{\text{friction}}$) and local losses (Δp_{local}).

$$\Delta p = \Delta p_{\text{friction}} + \Delta p_{\text{local}} \quad (10)$$

$$\Delta p_{\text{friction}} = \frac{1}{2} \rho v^2 \frac{l}{d_H} \lambda \quad (11)$$

$$\Delta p_{\text{local}} = \frac{1}{2} \rho v^2 \xi \quad (12)$$

In above equations ρ denotes the fluid density, $v = \frac{Q}{A}$ the mean flow velocity, l and d_H the length and the hydraulic diameter of the flow channel of the labyrinth, λ the friction factor and ξ the total coefficient of local resistance.

Frictional Losses

Frictional losses in a pipe are losses due to the viscosity η of the fluid as a result of the exchange of momentum between fluid molecules. At laminar flow viscosity forces are caused by the relative velocity of adjacent fluid layers. Due to the predominance of these forces, marginal roughness of the wall has no effect on the magnitude of the resistance and the friction coefficient is a function of the Reynolds number Re :

$$\lambda = \frac{c}{Re} \quad \text{with } Re = \frac{d_H \rho v}{\eta} \quad (13)$$

For circular cross sections the constant $c = 64$ can be determined according to the theory of Hagen-Poiseuille. For rectangular cross sections c is approximately in the range of $c \approx 70$. With increasing flow velocity inertia forces have to be considered and at the critical value of the Reynolds number Re_{lim} the flow becomes turbulent. For turbulent flow conditions, a number of different friction models exist depending on the wall roughness. In this work the approach according to Blasius for smooth pipes with circular cross section is used:

$$\lambda = 0.3641 \frac{1}{Re^{0.25}} \quad (14)$$

Equation (13) is valid for $Re < Re_{\text{lim}1} \approx 2400$ (laminar flow) and Equation (14) is valid for $Re > Re_{\text{lim}2} \approx 4000$ (turbulent flow). The transition zone between laminar and turbulent is often omitted in reference books. Hence, the friction factor is interpolated between $Re_{\text{lim}1}$ and $Re_{\text{lim}2}$ within this approach.