

Gerotor Pumps for Automotive Drivetrain Applications: A Multi Domain Simulation Approach

2011-01-2272

Published
09/13/2011

Wolfgang Schweiger and Werner Schoefmann
Magna Powertrain

Andrea Vacca
Purdue University

Copyright © 2011 SAE International

doi:10.4271/2011-01-2272

ABSTRACT

This paper presents a simulation model for the analysis of internal gear ring pumps. The model follows a multi domain simulation approach comprising sub-models for parametric geometry generation, fluid dynamic simulation, numerical calculation of characteristic geometry data and CAD/FEM integration. The sub-models are interacting in different domains and relevant design and simulation parameters are accessible in a central, easy to handle graphical user interface.

The potentials of the described tool are represented by simulation results for both steady state and transient pump operating conditions and by their correlation with measured data. Although the presented approach is suitable to all applications of gear ring pumps, a particular focus is given to hydraulic actuation systems used in automotive drivetrain applications. In particular, simulation results of entire drive cycles on the example of an electro-hydraulic actuated AWD coupling (considering electric control unit, electric motor, internal gear ring pump, pressure relief valve and multi-plate clutch) are presented in the final part of the paper.

The presented results aim to show the capability of the developed tool for design considerations: due to fully parameterized models, a comparison of different pump design concepts regarding to pump efficiency and system performance can rapidly be performed already in an early stage of the development process.

INTRODUCTION

The automotive industry is currently called on to develop new and innovative products and to reduce time effort, risk and cost of new developments at the same time. Increasing requirements regarding comfort, operating efficiency and emissions in combination with the augmented implementation of mechatronic systems in vehicle drivetrains gives need for new and comprehensive simulation solutions - both at component and system level.

Statements in terms of system efficiency characteristics over the entire range of operation are needed, already in an early stage of product development. Consequently, reliable predictions of sub- and auxiliary systems efficiency characteristics are mandatory and the predictive use of numerical tools is an essential part of the design and development process.

Internal gear ring pumps (GRPs) are the most commonly used pumps in automotive fluid power applications. Compared to other pump types they offer small package size, low manufacturing costs and can be used within a wide range of applications (e.g. fuel pumps, ICE lubricating oil pumps, oil pumps in automatic transmissions, electro-hydraulic power packs). However, despite these advantages, their operation is characterized by several complex phenomena of physical nature related to the features of the flow, to the peculiar geometry of the unit, to the displacement and to the deformations of the mechanical parts. The simultaneous presence of those aspects strongly complicate the analysis of GRPs, and reliable tools suitable for design purposes are currently not available. This makes the understanding of the effects due to the main design parameters (such as

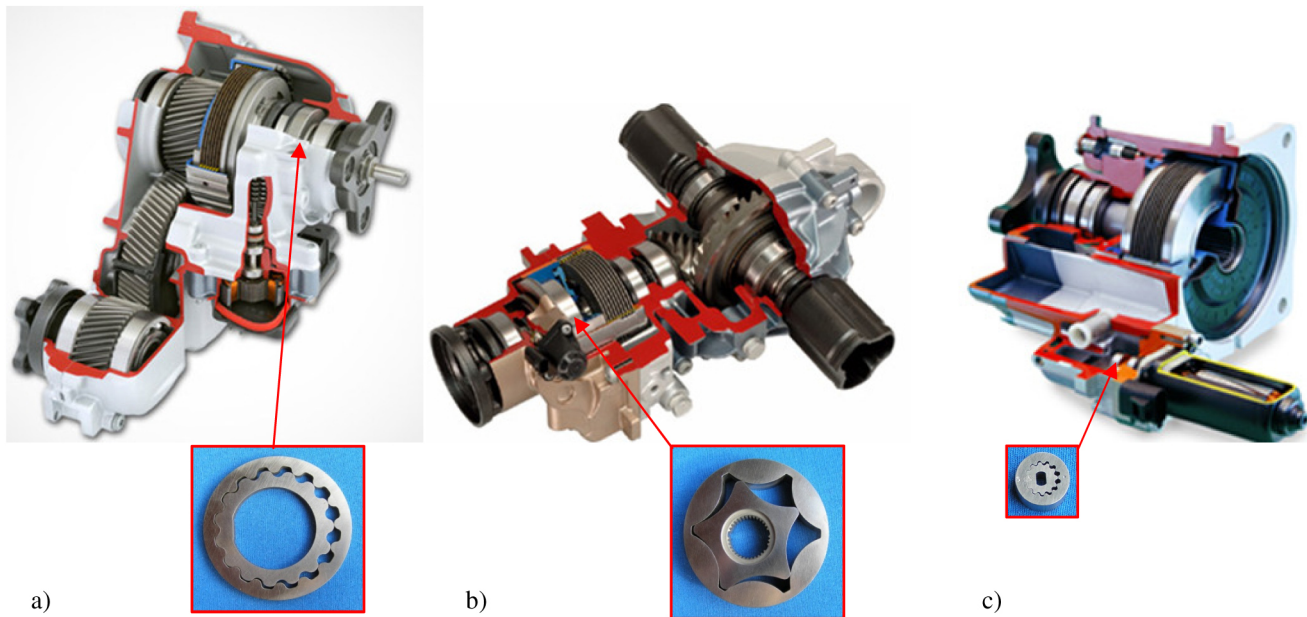


Figure 1. MAGNA Powertrain AWD-systems: a) Active Transfer Case (ATC) 350 with lubrication oil pump; b) PROACTIVE Coupling with differential speed sensing pump; c) DYNAMAX Coupling with electric motor-pump unit.

dimensions and tolerances) on pump operation not yet completed achieved.

The scope of this research work, which was initiated by MAGNA Powertrain, was the development of a numerical procedure for the design, analysis and optimization of GRPs, primarily for the use in torque control systems for automotive AWD drivetrain applications as shown in [Fig. 1](#).

Until now the process of designing and integrating GRPs has been mainly based on empirical procedures, strongly relying on experimental trial and error approaches and on the knowledge of experienced pump designers. Only for limited aspects of pump design, numerical tools recently became of frequent use (geometric analysis of GRP kinematics, FEM structural analysis, etc). As presented in the next section, several previous studies focused on the detailed analysis of particular effects characterizing GRP operation, but an overall approach for the complete simulation of the unit and of its dynamical interactions with a particular actuation system, e.g. an electro-hydraulic system as shown in [Fig. 2](#), has never been considered with high level of detail.

Existing design approaches difficultly permit to formulate an optimal design solution for GRP as a function of the application. Usually, to satisfy the requirements of most applications, boundary conditions for the pump design are predetermined by package size, required maximum pump pressure and flow rate. These information can be deduced out of system functional requirements which are in automotive applications typically characterized by a wide range of operating temperature, possible misuse operation (kick-off

start, towing) and high demands on system dynamics. Out of this, the need for a comprehensive system simulation, including a detailed physical pump model which considers dynamic interactions between the mechanical and the hydraulic domain, can be deduced.

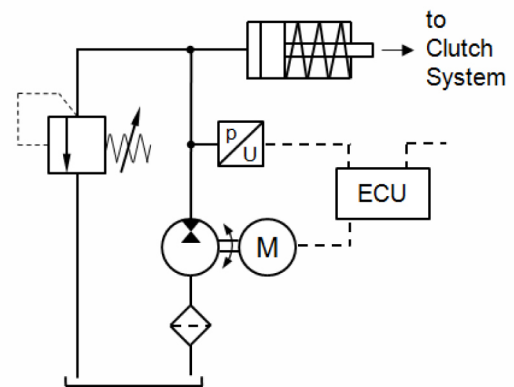


Figure 2. Hydraulic circuit of the electro-hydraulic actuated coupling shown in [Fig. 1c](#).

With this motivation, the main focus of the research is to define a simulation tool capable of identifying the main effects associated with the basic design parameters of GRPs. This has been realized through the implementation and interaction of sub-models in different simulation domains. Although the present work represents the first step towards the realization of an omni-comprehensive simulation model and few important aspects are not yet considered in the model, the results presented in this paper demonstrate how at

current state of development the model is already useful for design considerations.

The structure of the paper is as follow: after a brief introduction and classification of different types of internal gear ring pumps that are typically used in automotive drivetrain applications, the focus is given to the description of the modeling approach and of the main potentials of the developed tool in terms of a holistic system analysis. The different simulation parts, their associated sub/models and their interaction in different simulation domains are described. In the last part, first results based on the current stage of development, for different pump designs and operating conditions are presented and the results are compared to measured data. Finally, the research takes as particular reference the closed loop control system for the electro-hydraulic coupling of Fig. 1c, whose ISO hydraulic sketch is represented in Fig. 2. In this system, the pump is used for pressure supply and dynamic pressure control of the linear actuator, according to the current force demand.

LITERATURE OVERVIEW

The concept and the operating principle of GRPs as an alternative to reciprocating piston devices captivated the attentions of researchers since the beginning of the 20th century, as documented by F. Wankel, pioneer on the development of pistonless rotary displacement machines for industrial application. In his comprehensive classification of rotary piston machines in 1963 [1], Wankel documented also the efforts made in developing GRPs.

After Wankel's work, many studies have been conducted on GRPs, focusing on several aspects such as the features and the generation of gear teeth profiles with respect to the volumetric efficiency, sealing performance and outlet flow [2,3,4,5,6] or the analysis of the forces and moments acting on the gears [7].

Recently, significant effort has been put also on the development of numerical models suitable for GRP design purposes. In particular, Fabiani et al. [8] mainly focused on the geometric and kinematic modeling. Neyrat et al. presented in [9] a 1D AMESim® model that combines an advanced model for the evaluation of fluid properties with a geometrical model for GRP; a similar model was presented by Manco' and Nervegna in [10,11]. Following a similar approach, Gamez-Montero presented an in-house simulation package named GEROLAB [12]. A commercial simulation tool, Pumplinks, has also been made available for detailed CFD-based analyses of the flow through the GRP [13].

Although from one side the mentioned works accomplished the goal of improving the knowledge of the mechanisms involved in the operation of GRPs, on the other side the authors assumed proper simplifications (i.e. negligence of

interaction between the fluid dynamic and the mechanical domain, of the deformation effects due to fluid pressure, etc.) that permit to focus on a particular aspect or they developed numerical models which are not easily adaptable to parametric analyses of different designs.

APPLICATION OF GEROTOR PUMPS IN AUTOMOTIVE AWD SYSTEMS

As shown in Fig. 1 in drivetrain applications, typically three types of GRPs can be distinguished:

- **Absolute speed driven GRPs:** driven by crankshaft or cardan shaft absolute rotational speed. (usually within the range of 500 ... 9000 rpm).
- **Differential speed sensing GRPs:** driven by rotational speed difference of input and output shaft, proportional to slip e.g. occurring between front and rear axle speed (usually within the range of $-500 \dots +500$ rpm).
- **Electric motor driven GRPs:** driven by electric motor rotational speed, corresponding to the ECU control strategy; so-called electro-hydraulic power packs (usually within the range of 0 ... 6000 rpm).

Hydraulic actuated AWD systems, as shown in Fig. 1b,c provide torque control via a multi plate clutch. The transferred torque is nearly proportional to the pressure acting on the clutch piston. According to their operating principle they can basically be divided into hydromechanic actuated couplings containing differential speed sensing pumps (Fig. 1b) or electro-hydraulic actuated couplings, providing fully active torque control, independent from the current vehicle driving condition (Fig. 1c). A more detailed description of mentioned AWD systems is presented in [24]. The area of usage is usually limited to applications with 30 bar.

To describe the components and the basic operating principle of a GRP, the case of an electric motor-pump unit is shown in Fig. 3. The figure displays the different parts of this type of positive rotary displacement machine: the rotor set (1) with an inner rotor (2) and an outer rotor (3) and a housing comprising base plate (4), mid plate (5) and cover plate (6). The outer rotor has one more teeth than the inner rotor and is positioned at a fixed eccentricity e from the centerline of the inner rotor and shaft. As the rotors rotate about their respective axes, fluid is sucked into the inlet port (7) due to the enlarging chambers. With continuing rotation chamber volume decreases and fluid is transferred to the delivery port (8).

The electric motor-pump unit shown in Fig. 3 is taken into particular consideration in the results section of this paper, when the example of an electro-hydraulic actuated AWD system is considered. Despite this particular application taken as reference, the main aim of this paper is to show the

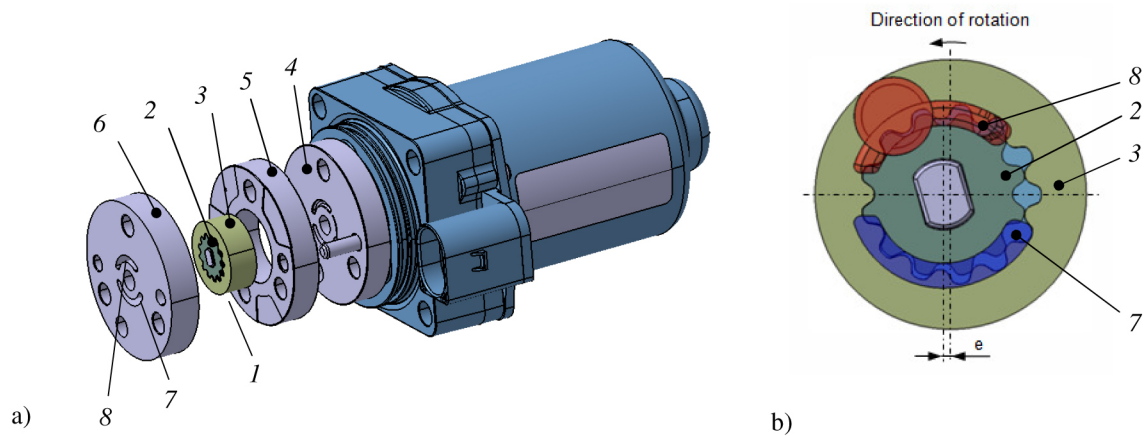


Figure 3. a) Exploded view of an electric motor-pump unit; b) Detail of rotor set with porting contour.

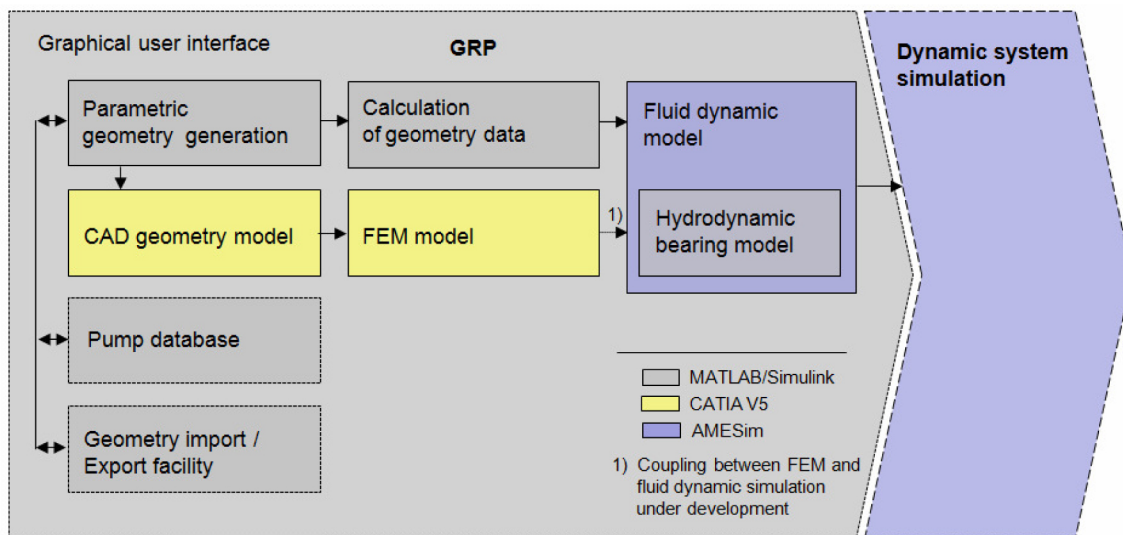


Fig. 4. Structure and interaction of the sub-models.

potentials of the developed simulation tool for generic GRP designs.

MODELING APPROACH

The proposed modeling approach is constituted of five basic models which are interacting in different domains. According to Fig. 4 it comprises models for parametric geometry generation and for numerical calculation of characteristic geometry data, a 3D CAD geometry model, a FEM-model and a lumped parameter fluid dynamic model. A model of the hydrodynamically lubricated outer rotor support is integrated within the fluid dynamic model.

The parametric geometry generation and the numerical calculation of characteristic geometry data which is required in further simulation steps, is performed within *Matlab/Simulink*. The fluid dynamic simulation is implemented within the *Imagine.Lab AMESim* simulation environment. The

CAD and FEM models are implemented in *CATIA V5* using the Part Design, Assembly Design, *DMU Kinematics* and the *Structural Analysis Workbench*.

The models and their interactions are controlled by a global graphical user interface (GUI), based on a *MATLAB* script. All essential pre- and post-processing operations can be performed out of the GUI, shown in Fig. 5. It enables access to relevant design- and simulation parameters (1), (2) which are used in different domains. The GUI includes a pump layout window (3) for 2D geometry visualization, configurable simulation result plot windows (4) and a pump library (5) containing parameter sets of existing GRPs which were analyzed in the framework of present study. Due to a geometry import feature (6), 2D CAD data as well as measured teeth profile coordinates can be imported. Furthermore, new pump design parameter sets can be managed within the database and data of tooth profiles as

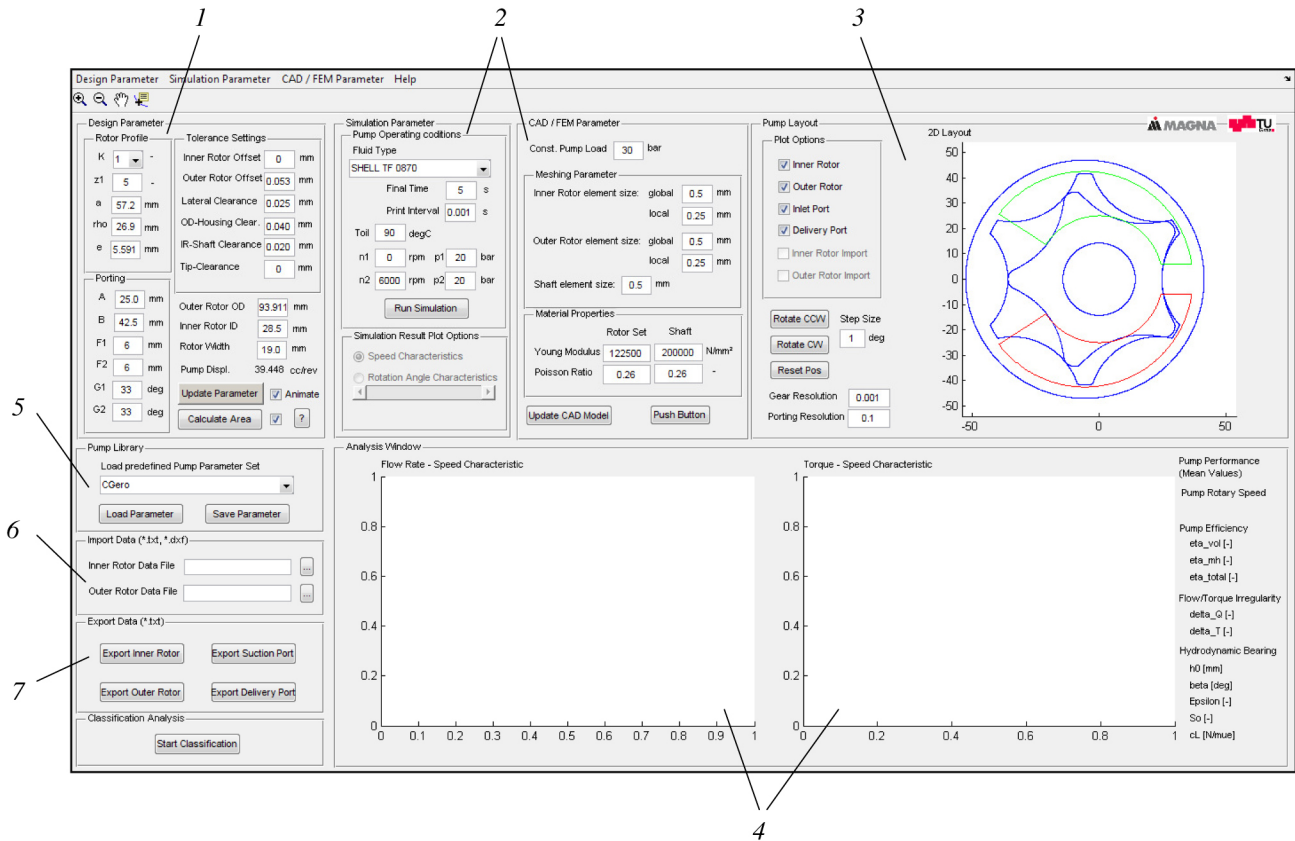


Fig. 5. Graphical User Interface.

well as in- and outlet porting geometry can be exported due to an export utility (7).

The pump layout window (Fig. 5) enables a graphical appraisal of the current parameters' setting; furthermore imported teeth profile data can be superimposed. This permits an easy graphical comparison of the current pump design with numerically generated profiles or with measured data / CAD data of existing rotor profiles, as it will be shown in the following result section.

GEOMETRY GENERATION

A geometric preprocessor generates the rotor and porting geometry, based on few basic design parameters as shown in Figs. 6 and 7. Different coordinate systems are used: system (1) defined by (O_1, x_1, y_1) and system (2) defined by (O_2, x_2, y_2) , rigidly connected to rotor 1 and rotor 2 respectively. The basic trochoidal curve is generated in coordinate system 1 by a point A that is rigidly connected to a circle of radius r_2 which is rolling, without sliding, along the inner or outer circumference of another fixed circle of radius r_1 . The geometry of the first rotor is an envelope of a trochoid, the second rotor is obtained as its conjugate and has a number of teeth that differs by one. The mathematical description of trochoidal gear profiles is described in detail in [4,5,6,7].

According to Bonandrini et al. [4], based on the basic law of gearing, the profiles of the trochoidal rotor and its conjugate can be expressed by original equations, and the geometry is completely defined by the choice of five geometrical parameters: K, z_1, a, ρ, e which are accessible by the design parameter tab of the GUI (Fig. 5).

The procedure for geometry generation involves the steps as described below:

1. Generation of a basis trochoidal curve based on analytical equations. This can either be epitrochoidal (if the rolling circumference is internal, $K = +1$) or hypotrochoidal (if the rolling circumference is external, $K = -1$) curves.
2. Determination of the first rotor profile as envelope of the basis trochoidal curve. The trochoidal rotor profile represents an equidistant to the basis trochoidal curve. This is performed in system 1 by means of a unit vector $\vec{n}$ normal to the trochoidal profile and the envelope radius ρ . According to a positive or negative value of ρ , either the internal ($\rho > 0$) or external ($\rho < 0$) envelope is obtained.
3. Numerically computation of the contact points and the conjugate profile for the second rotor, based on the theory of gearing. The unit vector $\vec{n}$ is the same for the conjugant

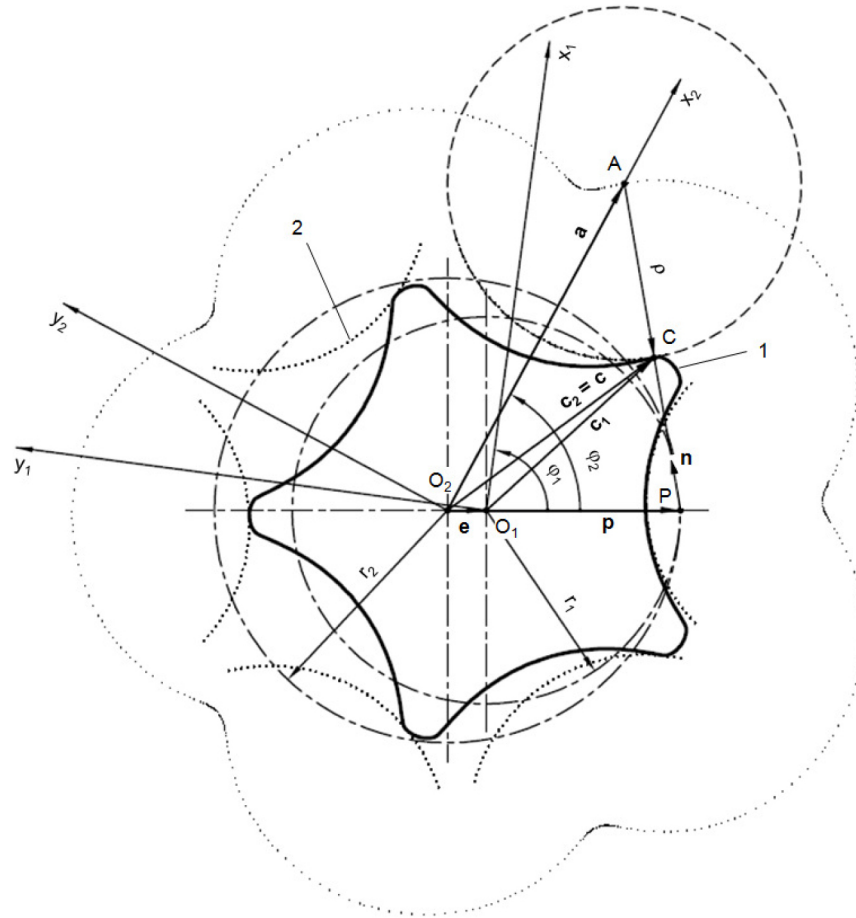


Fig. 6. Generation of trochoidal profiles.

profile and for the basis trochoidal profile and it contains the contact points.

Although, in accordance to Bonandrini et al [4], the classical design consist of the use of the epitrochoidal curves with internal envelope ($K = +1$, $\rho > 0$), hypotrochoidal curves with outer envelope ($K = -1$, $\rho < 0$) can be used as an alternative solution.

The numerical method of geometry generation and the corresponding equations are explained by use of a configuration of an epitrochoidal curve with an internal envelope as it is shown in Fig. 6.

The center points O_1 , O_2 of the subsystems (1), (2) as well as the instantaneous center of rotation P are fixed and arranged on a line. P is defined by the coordinates $\begin{bmatrix} r_2 \\ 0 \end{bmatrix}$, the vector $\vec{P}$ is pointing to P. Point of origin O_2 is coincident with the center of rotation of system (2). Vector $\vec{e}$ points to the center point of system (1), it is $r_2 - r_1 = \|\vec{e}\|$. $\vec{e}$ is defining the pump

eccentricity with the coordinates $\begin{bmatrix} e \\ 0 \end{bmatrix}$. The two circles of r_1 and r_2 are rolling, without sliding.

The basic trochoidal curve is traced by point A which is rigidly connected to circle a with a constant radius a and the angle ϕ_2 . Vector $\vec{a}$ is pointing to A. The relations are expressed by Eq. 1,2,3,4,5,6,7,8,9,10.

$$z_1 = z_2 + \frac{1}{2} \cdot (1 - K) \quad (1)$$

$$z_2 = z_1 + K \quad (2)$$

$$r_1 = \frac{e \cdot \frac{z_1}{z_2}}{1 - \frac{z_1}{z_2}} \quad (3)$$