

Modelling and Simulation of Brake Booster Vacuum Pumps

Massimo Rundo
Politecnico di Torino

Raffaele Squarcini
Pierburg Pump Technology Italy

ABSTRACT

Aim of this work is the development of a lumped parameters simulation model of single-vane vacuum pumps for pneumatically actuated brake boosters. Kinematic and fluid-dynamic models are integrated in a simulation environment to create a tool aimed at evaluating the vacuum pump performance and at guiding the designer during the prototype development. The paper describes extensively the mathematical model, the time domain simulation and experimental analyses performed on a camshaft mounted unit. Great emphasis is placed on the evaluation of the geometric quantities of the control volumes into which the vacuum pump has been divided. For each control volume the mass and energy conservation equations lead to the determination of the instantaneous pressure. The volume of each variable chamber and the respective angular derivative are calculated as function of the shaft position starting from the stator track profile supplied as a generic closed polyline. Flow areas between each chamber and the inlet/outlet volumes during a complete shaft revolution are evaluated directly from a data file containing the x-y coordinates of the passage area contour. Different leakage paths are also taken into account. Specific attention is dedicated to the simulation of the lubricating fluid throughout the vacuum pump in order to take into account possible overpressures due to the presence of a trapped volume of oil, above all during cold starts. To achieve this target the equations describing the behavior of the liquid and gaseous phases are applied simultaneously and the current fraction of oil is evaluated for each variable volume chamber. The model is able to work with an oil fraction ranging from 0% to 100% without introducing discontinuities in the differential equations. Experimental tests have confirmed a good matching with the simulation results in terms of pressure vs. time characteristics and absorbed torque.

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INTRODUCTION

In automotive braking systems, the force exerted by the driver on the foot pedal is amplified by the vacuum brake booster, a pneumatic servo system mounted between the pedal push rod and the tandem master cylinder. It is provided with a linear actuator that converts the pressure difference between its two chambers into a force proportional to the driver effort. The energy necessary to generate such servo-assistance force is supplied by a vacuum source. Two alternative solutions are possible: to exploit the engine intake manifold depression or to use a separated vacuum pump. This last component is used above all on Diesel engines, but also increasingly on gasoline engines. Furthermore the vacuum pump is also utilized as negative pressure source for other systems, such as the variable geometry turbocharger (VGT) actuator or the exhaust gas recirculation (EGR). As for other

engine ancillaries, in recent years due to the severe regulations for lowering the fuel consumption, the optimization of the vacuum pump is becoming of great relevance [1]. Time to market reduction can be achieved through a flexible simulation model able to appraise the influence of different geometric parameters on the unit performance. However the development of a lumped parameters model of an automotive vacuum pump is a challenging task, since such component works not only with air, but also with lube oil to improve the sealing between moving parts, to remove the heat and to reduce the friction. In most operating conditions the presence of the liquid phase could be disregarded, however at the engine start up the amount of oil could be so relevant that cannot be neglected. In such condition the vacuum pump behavior should be assimilated to a hydraulic machine rather than a pneumatic unit. Aim of the present paper is the development of a

comprehensive and as general as possible simulation model of a vacuum pump able to work with high oil volume fractions.

UNIT DESCRIPTION

A view of the vacuum pump under study is shown in [Figure 1](#). Three variable volume chambers are originated by a single vane sliding inside an eccentrically mounted rotor directly driven by one of the camshafts. Both ends of the vane are provided with a circular arc slipper always in sliding contact with the stator track. The stator profile is constituted by a suitable curve closed by a circular arc to form a sealing surface with the rotor between high and low pressure chambers. The main user (brake booster) and secondary user (e.g. VGT actuator) are connected with the suction side of the vacuum pump through hose fittings with an integrated non-return valve. The suction hole of the secondary user is located on the opposite side with respect to the primary inlet port, in order to avoid a direct connection between the users for any position of the vane. In this way, in case of failure of the secondary circuit, it will be still possible to generate the vacuum for the brake booster.

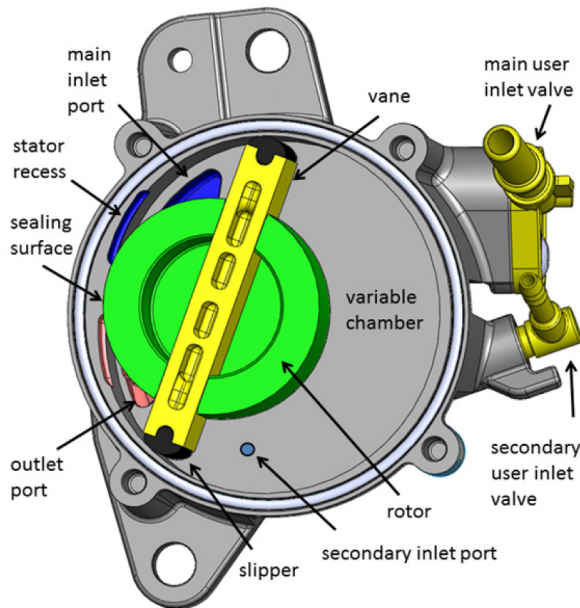


Figure 1. vacuum pump assembly

The oil coming from the engine lubrication circuit is supplied through the feed hole of the shaft bearing and the axial clearance between rotor and casing. The mixture of air and oil is discharged from the pump to the engine crank case through a reed valve located on the rear side ([Figure 2](#)). Two recesses on the stator track allow a direct connection between adjacent chambers in order to avoid trapped oil volumes.

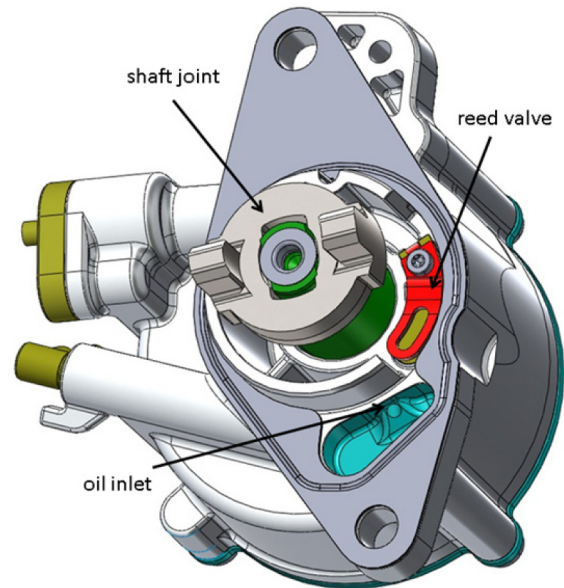


Figure 2. rear view

[Figure 3](#) shows the reference position of the vane, the positive direction of rotation, the numbering of the three variable chambers and of the contact points. The working cycle of a chamber is completed in a 540° rotation of the camshaft, hence the number of cycles per shaft revolution of each chamber n_c is equal to $2/3$. The displacement of the vacuum pump is given by the [Eq. \(1\)](#):

$$V = 3 \cdot V_0 \cdot n_c = 2 \cdot V_0 = 2 \cdot (V_{\max} - V_{\min}) \quad (1)$$

where the unit displacement V_0 is defined as the difference between the maximum and minimum volume of a chamber.

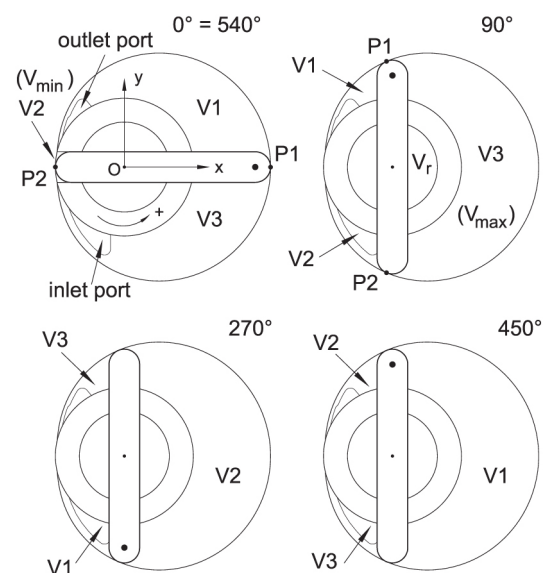


Figure 3. reference positions

GEOMETRIC MODEL

Vector Rays Evaluation

The internal stator track profile is defined by a curve L , generally supplied as a two-column table in the reference frame x, y with center O (Figure 4). The vector ray $\mathbf{r}$ identifies the trajectory of the contact point P along the curve L . To determine the vector ray module ρ and its phase ψ as function of the shaft angular position ϕ , the auxiliary reference frame u, v is defined, where the direction of the unit vector $\mathbf{u}$ is coincident with the vector ray. The derivative of the vector $\mathbf{r}$ is tangent to the curve L in the contact point P . In the reference frame u, v it is expressed as:

$$\mathbf{r}' = \frac{d\rho}{d\psi} \cdot \mathbf{u} + \rho \cdot \mathbf{v} = \overline{PK} \cdot \mathbf{u} + \overline{MK} \cdot \mathbf{v} \quad (2)$$

Thus its module is:

$$|\mathbf{r}'| = \sqrt{\left(\frac{d\rho}{d\psi}\right)^2 + \rho^2} \quad (3)$$

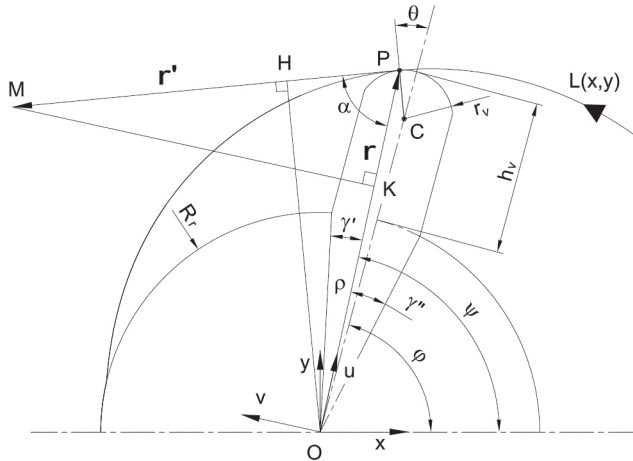


Figure 4. vector ray and reference frames

If the rectangular triangles PKM and OPH are considered, the following identities can be determined:

$$\overline{PH} = \rho \cos \alpha = \rho \frac{\overline{PK}}{\overline{PM}} = \rho \frac{d\rho}{|\mathbf{r}'|} \quad (4)$$

$$\overline{OH} = \rho \sin \alpha = \rho \frac{\overline{MK}}{\overline{PM}} = \frac{\rho^2}{|\mathbf{r}'|} \quad (5)$$

The vane lift can be expressed as:

$$h_v = \overline{OC} + r_v - R_r = \sqrt{(\overline{OH} - r_v)^2 + \overline{PH}^2} + r_v - R_r \quad (6)$$

Moreover the following angles can be determined

$$\alpha = \arctan \left(\frac{\rho}{\frac{d\rho}{d\psi}} \right) \quad (7)$$

$$\theta = \arctan \left(\frac{-\frac{d\rho}{d\psi} \cdot \rho}{\rho^2 - r_v \cdot |\mathbf{r}'|} \right) \quad (8)$$

The relationship between the angular position of the vane axis ϕ and the phase of the vector ray is given by the Eq. (9):

$$\phi = \psi + \alpha - \theta - \frac{\pi}{2} \quad (9)$$

Since the angular position of the vane ϕ is the input variable, the inverse functions $\rho(\phi)$, $\psi(\phi)$, and $h_v(\phi)$ are evaluated numerically by linear interpolation.

Chambers Volume Variation

A variable volume chamber is delimited by two vector rays p_j and p_{j+1} as shown in Figure 5. Depending on the angular position of the shaft, a vector ray must be considered fixed with module equal to the rotor radius or rotating. Three different configurations must be considered during the entire working cycle of a chamber:

$$\begin{cases} \rho_j = \rho(\phi) & \rho_{j+1} = R_r & \text{if } 0 \leq \phi < \pi \\ \rho_j = R_r & \rho_{j+1} = \rho(\phi) & \text{if } \pi \leq \phi < 2\pi \\ \rho_j = \rho(\phi - \pi) & \rho_{j+1} = \rho(\phi - 2\pi) & \text{if } 2\pi \leq \phi < 3\pi \end{cases} \quad (10)$$

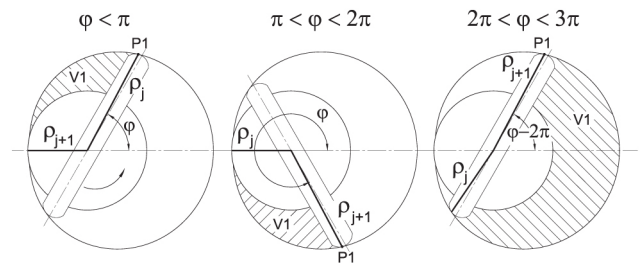


Figure 5. vector rays delimiting the chamber 1

The angular derivative of a chamber can be evaluated directly from the vector rays and vane lift derivatives with the Eq. (11) [2]:

$$\frac{dV_c}{d\phi} = \frac{1}{2} H \left[\rho_{j+1}^2 - \rho_j^2 - 2R_r \left(\gamma'_{j+1} \frac{dh_{v,j}}{d\phi} + \gamma''_{j+1} \frac{dh_{v,j+1}}{d\phi} \right) \right] \quad (11)$$

The chamber volume V_c is evaluated by numerical integration of the derivative with the addition of the volume in correspondence of the reference position; this last term is calculated by integration of the area swept by the vector ray minus the volume V_{v0} occupied by half a vane:

$$V_c = \int_0^\varphi \frac{dV_c}{d\varphi} d\varphi + \frac{1}{2} H \int_0^\pi (\rho^2(\varphi) - R_r^2) d\varphi - V_{v0} \quad (12)$$

Flow Areas

Each chamber is connected alternatively with the inlet and outlet ports through the port plate. The actual flow area is given by the intersection between the frontal surface of the chamber and the contour of the port. It is evaluated by means of a graphical-analytical procedure that allows to calculate the area in common between two overlapping polygons. With reference to Figure 6, a 2 column matrix containing the X-Y coordinates of a closed polyline defining the profile of the chamber (curve A) is automatically constructed for different rotor positions with an angular step of 1° . A further 2 column matrix is built for the port profile (curve B) directly from the CAD drawing by using an AutoLISP routine or, as alternative for simple geometries, starting from the port parameters (step 1). A rectangular region containing the common area is determined by finding the maximum and minimum coordinates of the curves A and B (step 2) as follows:

$$\begin{cases} x_{\min} = \max(\min(x_{A,i}), \min(x_{B,i})) \\ x_{\max} = \min(\max(x_{A,i}), \max(x_{B,i})) \end{cases} \quad (13)$$

In such region a grid of $n \times m$ elements is constructed. Step 3 consists in identifying the elements whose center P is located inside the curve A. The generic point P is considered enclosed if an odd number of intersections between a vertical vector from P and the closed polyline is detected.

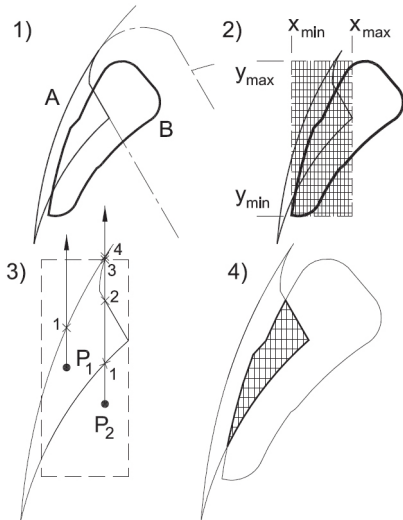


Figure 6. method for the evaluation of the flow area

For instance, the point P_1 is considered inside since there is only one intersection, while the point P_2 outside since the vector intersects the polyline four times. Only for the points inside the curve A, a second similar test is executed to determine if they also belong to the area delimited by the curve B (step 4). The resulting number of elements enclosed inside both curves multiplied by the unit area supplies the flow area (see the Appendix for CPU times and errors). The connection between each chamber and the secondary user line is realized by a circular hole (see Figure 1), whose diameter and center coordinates are supplied by the user as parameters. In this case the flow area can be calculated analytically as the surface of a circular segment progressively covered or uncovered by the vane. Each chamber can also communicate with the other two chambers through the recesses realized on the stator track. Based on the angular position of the vane (Figure 7), when the phase of the vector ray is $\pi - \chi_b < \psi < \pi - \chi_e$ the minimum distance d_r between the slipper tip and the edges A and B is calculated as follows:

$$d_r = \begin{cases} \sqrt{(x_A - x_C)^2 + (y_A - y_C)^2} - r_v & \text{if } \psi \leq \pi - \frac{\chi_b + \chi_e}{2} \\ \sqrt{(x_B - x_C)^2 + (y_B - y_C)^2} - r_v & \text{if } \psi > \pi - \frac{\chi_b + \chi_e}{2} \end{cases} \quad (14)$$

The coordinates of the center of the vane tip are evaluated starting from the vane lift:

$$\begin{cases} x_C = (h_v + R_r - r_v) \cdot \cos \varphi \\ y_C = (h_v + R_r - r_v) \cdot \sin \varphi \end{cases} \quad (15)$$

Hence the flow area, called h_r , the axial height of the recess, is evaluated as:

$$A_c = h_r \cdot \min(d_r, b_r) \quad (16)$$

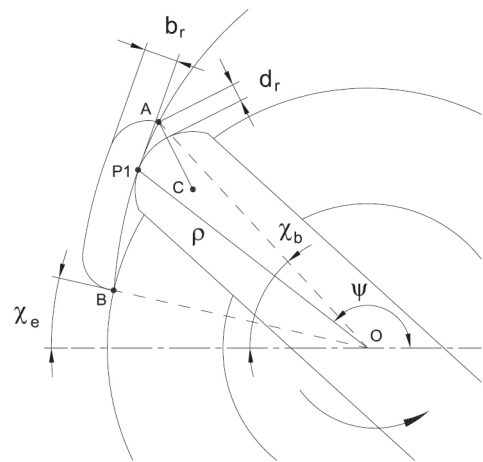


Figure 7. calculation of flow area on stator track