

Piloted Displacement Controls for ICE Lubricating Vane Pumps

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ABSTRACT

The paper brings to evidence critical issues in the design of variable displacement vane pump controls for ICE lubrication. Tight packaging, high rotational speed and variable pressure setting are key aspects influencing the selection of pump geometric parameters as well as its displacement controls.

A specific layout, where the pump displacement control is carried out by two linear actuators and a two-way pilot control valve, has been analysed.

The paper focuses on the dimensioning of the front areas of the actuators considering two kinematic solutions, linear and rotational, for the external ring. Through a simulation model, validated by experimental tests, the influence of different pressure control strategies on the dimensioning has been investigated.

INTRODUCTION

In recent years studies and research activities on variable displacement lubricating pumps for automotive applications have attracted a rapidly growing interest as research papers [1], [2], [3], [4] and numerous patents indicate.

Pump displacement controls may be conceived as either direct acting or as piloted solutions.

In [5] the effects of the external ring kinematics on the performance characteristics of vane pumps with direct acting controls were analysed.

These kinds of controls are normally appropriate in those applications where, at maximum engine speed, no risks exist of incomplete filling; in fact, rotor size and porting geometry optimization can contribute in preventing that risk while in more critical situations multi-stage unit designs can

also be considered and appraised. This notwithstanding, packaging limitations are often so stringent that the attained pump design becomes inadequate for high speed operation, thus leading to incomplete chambers filling [6].

To overcome this drawback piloted displacement controls should then be considered and particularly so if variable pressure settings are required within a control strategy of the lubricating network.

With reference to Fig. 1, the present paper considers an hydraulic layout where pump displacement control is carried out by two linear actuators with different front end areas. The smaller (*contrasting actuator* with area a) is subject to pump delivery pressure p_m . The larger (*displacement actuator*, area A) is instead acted upon by a controlled pressure p_x originated from delivery pressure. Pressure control is implemented by a remotely piloted valve (RPV) and a fixed orifice (FO) [7].

Restraints on the system follow:

- front end areas have an upper bound due to space restrictions, mainly in the axial direction in crankshaft mounted pumps;
- preload of the spring inside the larger actuator has lower and upper bounds: the force exerted must be high enough to keep the pump at maximum displacement at engine start, without limiting the minimum value of the pressure setting attainable by the displacement control;
- the remotely piloted valve and the orifice must guarantee pressure control in a sufficiently broad range.

Then the need exists for finding appropriate combinations of areas and spring that lead the pump to operate correctly in the expected operating range and meet the particularly stringent packaging constraints.

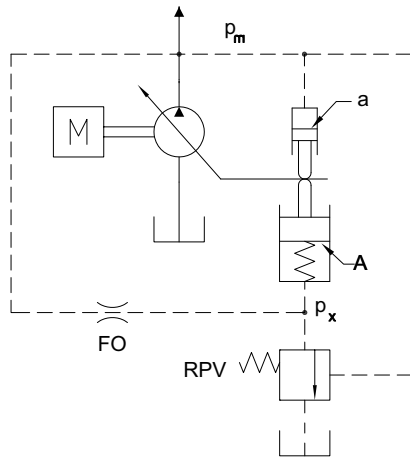


Figure 1: vane pump hydraulic scheme

Knowledge of the temperature dependent flow-pressure lubrication network characteristic is of prime importance to identify the most critical operating points.

Through a simulation model, the layout shown in Fig. 1 will be analysed in different operating conditions, considering two kinematic solutions (linear and rotational) for the external ring and different front end areas for the two actuators.

SIMPLIFIED THEORETICAL MODEL

EXTERNAL RING EQUILIBRIUM - Fig. 2 shows two pump schematics where the external ring can rotate about a planar joint with centre O_3 positioned in the suction (I) and the delivery (II) side of the unit. Actuators and spring forces have an arm length l_1 with the joint. At high speed, when incomplete filling is likely to occur, the pressure distribution on the inner circular face of the ring generates a perturbing resultant force [6]. In fact, if upon connection with delivery a chamber is not completely filled, a significant backflow occurs that eventually leads to a rise in pressure.

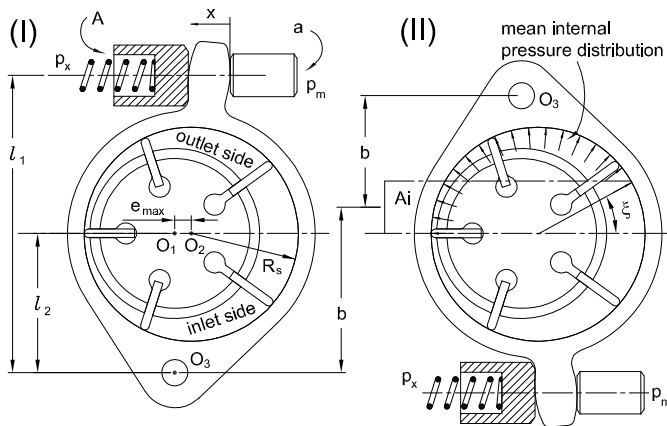


Figure 2: forces acting on the external ring

The internal pressure acting on an equivalent surface A_i develops a component force parallel to the actuators centerline that tends to reduce displacement. Its arm b can be

written as:

$$b = l_2 + i_c \frac{R_s}{2} \sin \xi \quad (1)$$

being i_c equal to +1 or -1 depending on the location of the joint, i.e. at the suction or, respectively, at the delivery side. As defective filling augments with speed, the angle ξ also increases. In displacement control conditions, the external ring equilibrium is written as:

$$p_m a l_1 + p_m A_i b = p_x A l_1 + (F_0 + kx) l_1 \quad (2)$$

If the system operates correctly, rotation may occur only if the controlled pressure p_x is lower than delivery pressure; this implies that the remotely piloted valve is also in control (not closed) and that the delivery pressure is equal to the RPV setting p^* . If an electrohydraulic proportional valve is used instead of the RPV, the pressure setting may be variable within low and upper limits, p_L^* and p_H^* respectively.

Introducing the following non dimensional quantities:

- pressure ratio $\Pi = p_x / p_m$
- displacement ratio $\alpha = e / e_{max}$
- arms ratio $\tau = l_1 / l_2$

equation (2) may be written as:

$$\Pi = \frac{1}{A} \left[a + A' - \frac{F_0 + \tau k (1 - \alpha) e_{max}}{p^*} \right] \quad (3)$$

where:

$$A' = \frac{A_i b}{l_1} \quad (4)$$

The case of linear motion (translation) of the external ring can be analysed by putting $A' = A_i$ and $\tau = 1$.

It is observed that pressure ratio depends not only on geometric parameters but also on operating conditions e.g. pump displacement and engine speed.

The pressure ratio Π given by eq.3 must be physically attainable by the system and may take the following extreme values $\Pi_L > 0$ with a completely open valve and $\Pi_H < 1$ when the valve is almost completely closed. Accordingly, if speed, pressure and current displacement are committed, parameters in eq. (3) must be so selected to comply with the condition $\Pi_L < \Pi < \Pi_H$.

Based on eq. (3), the most critical situation, when the pressure ratio Π reaches its maximum, occurs at:

- maximum speed (ω_{max}): worst situation for filling with maximum ξ and A_i ;
- maximum displacement ($\alpha = 1$): spring force is at minimum (minimum value of numerator);
- if variable, maximum control pressure ($p^* = p_H^*$): spring force becomes less relevant than the pressure induced force (maximum value of denominator).

However while the engine is running, the above situations are never coexistent because at maximum speed the pump operates at maximum pressure but at fractional displacement.

Therefore, an initial design specification is:

$$\frac{1}{A} \left[a + A'(\omega_{max}) - \frac{F_0 + \tau k(1 - \alpha_{max})e_{max}}{p_H^*} \right] \leq \Pi_H \quad (5)$$

being $\alpha_{max} < 1$ the displacement ratio at maximum engine speed.

The most critical situation, when pressure ratio reaches its minimum, occurs at:

- very low displacement: the spring is heavily compressed and applies its maximum force. In addition filling is complete and consequently $A' = 0$;
- if variable, minimum control pressure ($p^* = p_L^*$): spring force is important in regard to pressure force.

The above hypotheses lead to a second design specification:

$$\frac{1}{A} \left[a - \frac{F_0 + \tau k(1 - \alpha_{min})e_{max}}{p_L^*} \right] \geq \Pi_L \quad (6)$$

being α_{min} the minimum displacement ratio attainable by the pump.

ACTUATORS AREAS - For a given value of A eq. (5) and eq. (6) permit the evaluation of a range of values for a , where pump displacement is controlled in extreme situations and consequently in all intermediate conditions; the largest value of surface a_H in the contrasting actuator is:

$$a_H = A\Pi_H - A'(\omega_{max}) + \frac{F_0 + \tau k(1 - \alpha_{max})e_{max}}{p_H^*} \quad (7)$$

and the smallest a_L is:

$$a_L = A\Pi_L + \frac{F_0 + \tau k(1 - \alpha_{min})e_{max}}{p_L^*} \quad (8)$$

Fig. 3 shows a_H and a_L as functions of A . The couple of values a and A must fall inside the hatched triangular region. Below the value A_{min} no solutions exist; the higher is A beyond A_{min} the larger becomes the domain of feasible solutions for a . Within this domain, the choice of A and a influences the range of controlled pressure.

More specifically, for a given A , use of a_H for the area a , implies that, in the most unfavourable condition, one obtains $\Pi = \Pi_H$ i.e. the valve is on the verge of being closed; instead use of a_L , implies $\Pi = \Pi_L$ i.e. the valve is fully open.

To engender a safety margin it is advisable to keep pressure ratio variations within a narrower range (cross-hatched in Fig. 3) than the maximum allowed by the valve, that is:

$$\Pi_L + \Delta\Pi \leq \Pi \leq \Pi_H - \Delta\Pi \quad (9)$$

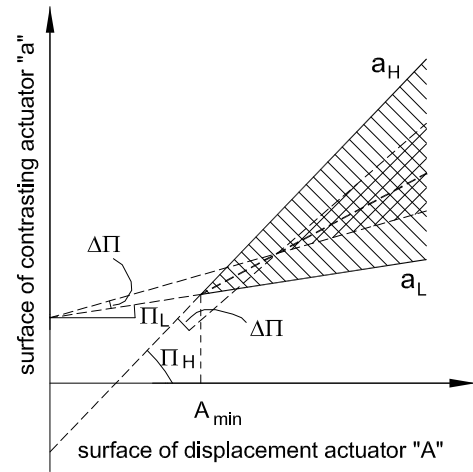


Figure 3: boundaries for actuators areas

This suggests selections as far away as feasible from the two straight lines representing a_H and a_L .

Therefore, for a given A it will be expedient to use the mean value between a_H and a_L (dashed line). However, as shown in Fig. 3, use of a narrower range for Π requires larger actuators areas.

SIMULATION MODEL

MODEL LAYOUT - For a precise appraisal of the internal force on the external ring, a dynamic lumped parameters simulation model, developed in AMESim™, has been used. The model, shown in Fig. 4, uses components belonging to a proprietary library developed at the Fluid Power Research Laboratory of the Politecnico di Torino (FPRL) for the accurate simulation of fixed and variable displacement lubricating pumps [8].

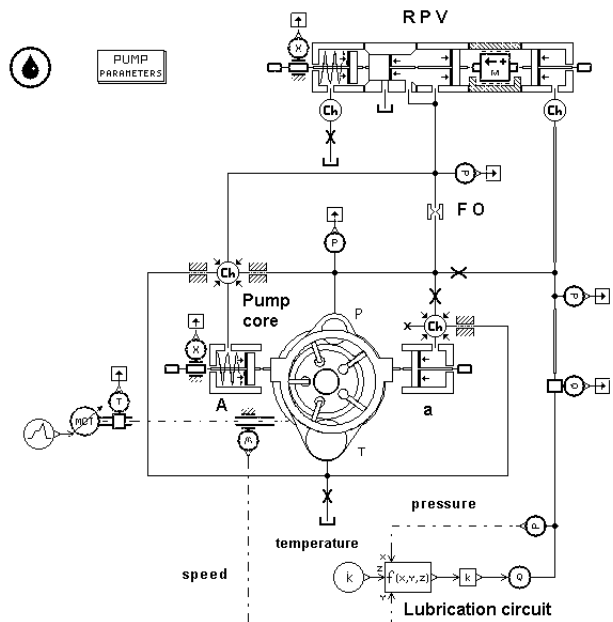


Figure 4: variable displacement pump simulation model