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APPLICATION OF CONTROL THEORY TO THE IMPROVEMENT OF A VANE PUMP FOR FASTER DYNAMIC RESPONSE

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ABSTRACT

Pump dynamic response, the speed in which a pump can build up pressure due to disturbance, system flow demand, is one of its most important characteristics. In high performance hydraulic control systems, normally we utilize a fast response, but expensive, variable displacement pressure compensated Piston Pump. Inexpensive Vane Pumps cannot be used because their very poor dynamic behavior, which is inherent from their construction and principle of operation. In this paper we apply Control Theory to analyze and investigate vane pumps and consequently suggest a possible re-design to improve their dynamic response. Performing stability, disturbance and speed of response analysis reveals that pump dynamic behavior is dominated by the parameter ratio Bvf / Ksp (Bvf– viscous coefficient, Ksp– spring constant). Upon completion of our study, we carried out a quantitative calculation of those parameters for a specific re-designed vane pump. In the re-designed vane pump speed of response was improved by factor of 35 and can now be used in high performance hydraulic servo systems, where high bandwidth is required.

KEY WORDS

Vane, Pump, Control, Dynamic, Response

NOMENCLATURE

M	:	Regulating ring mass	(Kg)
R	:	Regulating ring radius	(m)
Bvf	:	Viscous coefficient	(N / rad / sec)
Ksp	:	Spring constant	(N / m)
Kpv	:	Pressure coefficient	(N / bar)
Ke	:	Flow coefficient	(lpm / rpm / m)
K1k	:	Leakage coefficient	(lpm / bar)
N	:	Pump speed of rotation	(rpm)
δ	:	Ring – Plate clearance	(m)
ξ	:	Pump damping ratio	dimension-less
ωn	:	Pump natural frequency	(rad / sec)
Pop	:	Operating pressure	(bar)
Pdp	:	Desired / Set pressure	(bar)
Qp	:	System flow demand	(lit / min)
Cc	:	Oil compliance	(lit / bar)

INTRODUCTION

The introduction of the variable displacement vane pump dynamic model is a significant contribution to the field of hydraulic pumps. It enabled the control of pump performance characteristics via theoretical trade offs during the design process. We studied and analyzed this model relating to the Bvf / Ksp ratio, found the lower and upper limits of the spring constant Ksp and defined best possible value for viscous friction Bvf, for a faster pump. The study included building of the necessary block diagrams, derivation of the relevant transfer functions in S domain and applying Routh Criteria, Steady State and Dynamic analyses.

The results of the study of pump parameters, related to a specific pump, are as follows:

- Bvf has been decreased significantly
  Bvf new = 1 / 20 Bvf old
- Ksp has been increased (stiffer spring)
  Ksp new = 1.75 Ksp old

The newly designed pump shall have the following parameters, which dominate dynamic response:

Bvf / Ksp new = 1 / 35 Bvf / Ksp old

The numerical values are a result of the re-design of the specific vane pump used in a previous research, which also contributed the pump nonlinear block diagram.

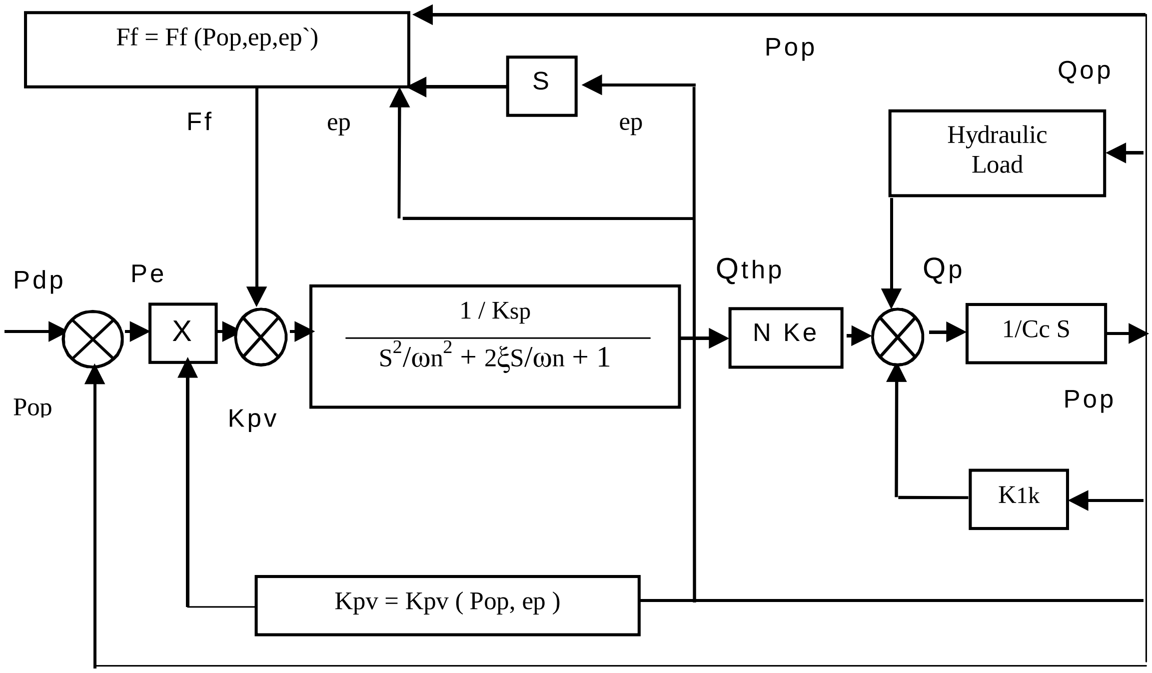

Figure 1: Pump nonlinear block diagram

STABILITY AND DISTURBANCE ANALYSIS

In order to study pump stability, we shall perform a linearized analysis of pump parameters neglecting the nonlinearities of the previous block diagram. The pump Pop/Pdp(s) transfer function derived from the linearized block diagram has the form (see figure no. 2):

$$\frac{Pop}{Pdp} = \frac{KpvNKe}{(MS^2 + BvfS + Ksp)(CcS + K1k) + KvpNKe} \qquad (1)$$

which we shall put also in the standard form and apply Routh Stability Criterion for stability analysis.

Pump regulating operation is a feedback system, with the following linearized block diagram:

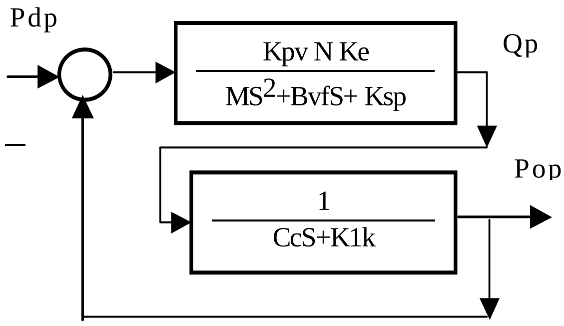

Figure 2: Pump linear block diagram with Pdp input

Pop / Pdp(s) in the standard form, equation no. (2):

$$\frac{Pop}{Pdp} = \frac{\dfrac{Kpv\ N\ Ke}{KpvNKe + KspK1k}}{[\dfrac{MCc}{KpvNKe + KspK1k}]S^3 + [\dfrac{BvfCc + MK1k}{KpvNKe + KspK1k}]S^2 + [\dfrac{BvfK1k + KspCc}{KpvNKe + KspK1k}]S + 1}$$

From the Routh Criterion we obtain the condition for stability, i.e. the constraint on spring constant Ksp:

$$(Bvf\ Cc + M\ K1k)\ (Bvf\ K1k + Ksp\ Cc) - M\ Cc\ (Kpv\ N\ Ke + Ksp\ K1k) > 0 \qquad (3)$$

since Bvf Cc^2 # 0, we solve for Ksp in equation no.(3) to get the following condition, equation no. (4):

$$Ksp \rangle \frac{M\ Cc\ Kpv\ N\ Ke - Bvf^2\ Cc\ K1k - M\ Bvf\ K1k^2}{Bvf\ Cc^2}$$

This mathematical condition provides the relationship between the spring constant Ksp and the other pump parameters, for a stable pressure compensated pump. From the equation, maximum pump natural frequency can be obtained, dividing by Bvf Cc^2 and neglecting the elements with minus sign to get the Ksp / M ratio:

Ksp / M $= \omega n^2 >$ Kpv N Ke / Bvf Cc and from here:

$$\omega n\ max = (Kpv\ N\ Ke / Bvf\ Cc)^{½} \qquad (5)$$

Let us turn now to the pump Steady State analysis. The steady state gain, from equation no. 2:

$$Gss = Kpv\ N\ Ke / (Kpv\ N\ Ke + Ksp\ K1k) \qquad (6)$$

or

$$Gss = 1 / [1 + (Ksp\ K1k / Kpv\ N\ Ke)] \qquad (7)$$

Since K1k, Kpv, N, Ke are given for a specific pump, steady state gain is inversely proportional to Ksp (spring constant) i.e. Gss α 1 / Ksp. Softer spring means higher steady state gain and better pump pressure regulation. Dynamic point of view, however, indicates opposite requirement namely a stiffer spring, because too soft a spring (equation no. 4) will cause system instability. The conclusion is that the spring constant has to be large enough to ensure stability, and small enough to have a good pressure regulation, a tradeoff must be performed. Another useful transfer function is pump pressure / disturbance. When a pressure compensated pump is operating under normal conditions, the desired input Pdp the (desired / set pressure) is usually kept constant. Pump dynamics is exercised by "disturbing" Qp flow demand of the system supplied by the pump.
The relevant linear block diagram with Qp as input:

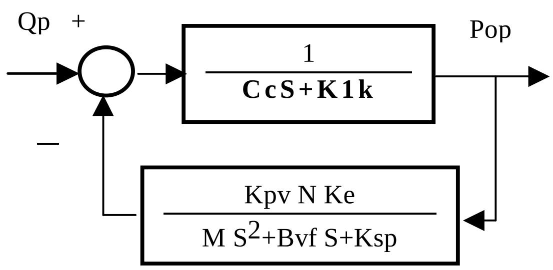

Figure 3: Pump linear block diagram with Qp input

The transfer function Pop / Qp (s) is obtained from the block diagram, equation no. (8):

$$\frac{Pop}{Qp}(S) = \frac{-(MS^2 + Bvf\ S + Ksp)}{(M\ S^2 + Bvf\ S + Ksp)(Cc\ S + K1k) + KvpNKe}$$

Or in its standard form, equation no. (9):

$$\frac{Pop}{Qp} = \frac{[\dfrac{-Ksp}{KpvNKe + KspK1k}]\ (\dfrac{S^2}{\omega n^2} + \dfrac{2\xi S}{\omega n} + 1)}{[\dfrac{MCc}{KpvNKe + KspK1k}]S^3 + [\dfrac{BvfCc + MK1k}{KpvNKe + KspK1k}]S^2 + [\dfrac{BvfK1k + KspCc}{KpvNKe + KspK1k}]S + 1}$$

where ωn^2 = Ksp / M is pump natural frequency, and
$2\ \xi / \omega n =$ Bvf / Ksp is viscous damping ration.
Routh Criterion, is identical to previous study of Pop / Pdp (s), so no additional information can be obtained.
The steady state gain Gss = - 1 / (KpvNKe/ Ksp + K1k) is directly proportional to Ksp, or Gss = α Ksp since again K1k, Kpv, N, Ke are given for a specific pump. From disturbance considerations a softer spring gives smaller pressure changes, on the other hand will cause to instability, so we see again trade off is necessary.

DYNAMIC RESPONSE ANALYSIS

Having applied Routh Criteria and performed stability and disturbance analysis we found the constraints on the spring constant Ksp, and realized that there must be a tradeoff between soft spring for better pump regulation and smaller disturbances, and stiff spring for stability. We now turn to the analysis and investigation of the other parameter of importance Bvf, the viscous friction coefficient, in order to find ways for its improvement.

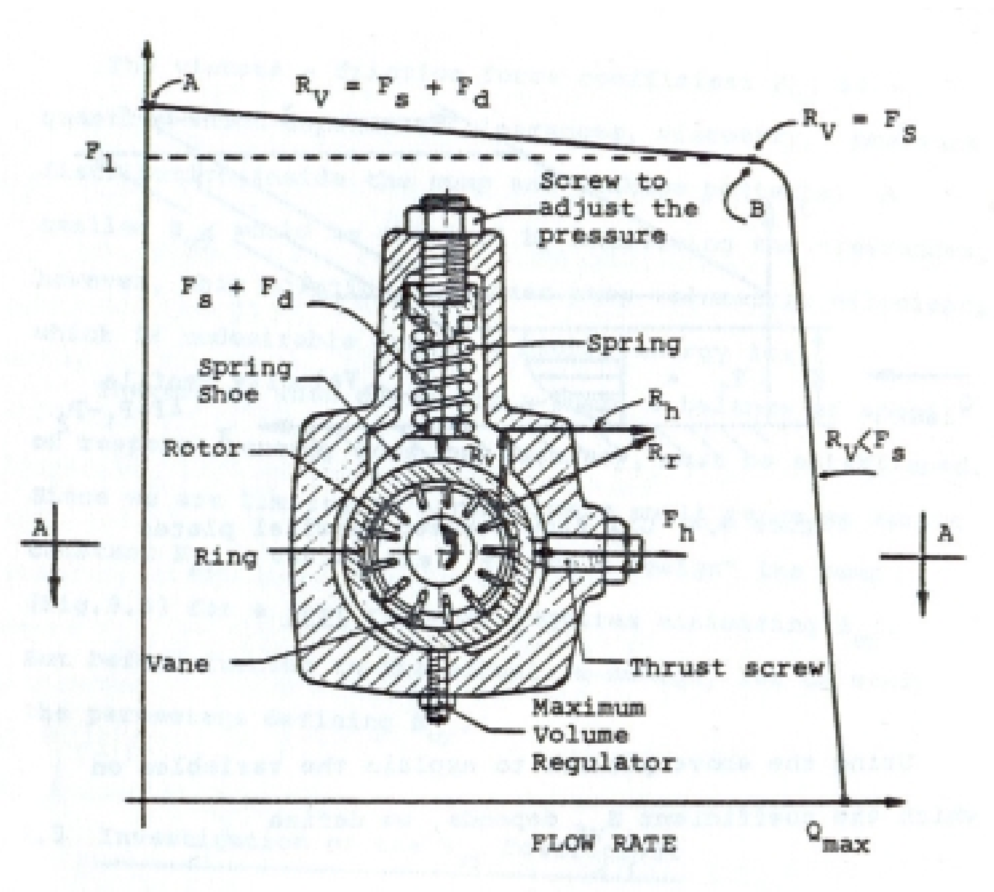

Figure 4: Typical variable displacement vane pump

Again, by quick pump response we mean "fast" pressure build up due to system flow demand Qp (flow disturbance). For a given operating point pump dynamic behavior is defined by the values of M, Bvf and Ksp, dominated by the ratio Bvf / Ksp,. For a typical pump Bvf / Ksp is in the range of 10^{-2} (Bvf / Ksp = 0.05), and the ratio M / Ksp =1 / ωn^2 in the range of 10^{-6} (ωn = 1068 rad / sec), the other parameters being constant. Pump Pop / Qp (s) transfer function can therefore by simplified to a second order system in the denominator, and a first order system with a time constant Bvf / Ksp in the nominator, equation no. (10).

$$\frac{Pop}{Qp} = \frac{-[\frac{Ksp}{KpvNKe + KspK1k}]\ (\frac{Bvf}{Ksp}S+1)}{[\frac{BvfCc + MK1k}{KpvNKe + KspK1k}]S^2 + [\frac{BvfK1k + KspCc}{KpvNKe + KspK1k}]S+1}$$

This transfer function is usually used in the literature as the mathematical model for pump dynamic behavior.
For a fast dynamic response the nominator time constant Bvf / Ksp has to be small, i.e. Bvf small and Ksp large.

INVESTIGATION OF THE Bvf PARAMETER

We assume that the Kpv, N, Ke, Cc, K1k parameters are not controllable by the designer, and we can't choose a stiff spring constant Ksp, due to stability constraints. The Bvf parameter is a function of the oil viscosity, ring clearances and leakage patterns. Smaller Bvf can be obtained by increasing clearances, however this would decrease volumetric efficiency and increase energy losses. We shall concentrate now on minimizing Bvf.

$$Bvf = \mu\ l\ h / \delta \qquad (11)$$

It is directly proportional to gap / friction surface, (l x h) and inversely to gap clearance δ, therefore to get smaller Bvf we need bigger gap clearances and less ring friction
The flow between the plates, leakage in our case,

$$Qleak = [(h\ \delta^3) / (12\ \mu\ l)]\ \Delta p \qquad (12)$$

Is directly proportional to the pump pressure drop Δp and sensitive to clearance δ^3, meaning bigger gap higher leakage. On the other hand we need bigger gaps for smaller Bvf, definitely contradicting requirements.
Refering to pump construction to define the parameters in Qleak and Bvf equations, let us look at pump cross section in figure 5. Ring is "pressed" between two plates

which seal the pumping chambers. Leakage Qleak is proportional to pressure drop Δp between Delivery port (High pressure) to Suction port (Low pressure).

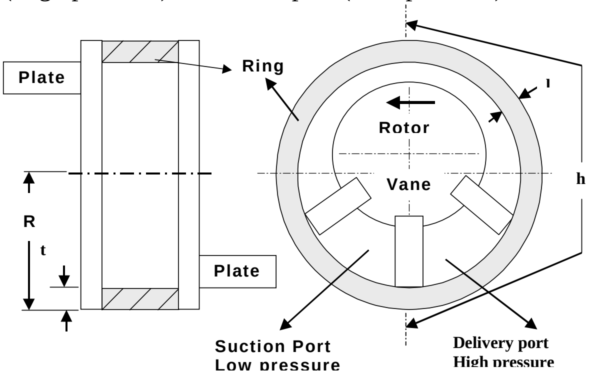

Figure 5: Pump ring – rotor configuration

Explanation of parameters l, h, δ: dimensions h~π R and l = t, in Bvf parameter, friction surface h x l = 4 π R l. The conclusion is that Bvf is large because the friction surface 4 π R l is large and gap is small (~ zero), since as mentioned earlier ring is "pressed" between plates, an inherent construction in vane pumps.

THE NEW PUMP DESIGN

Having performed qualitative and quantitave analyses of parameters Bvf and Ksp, affecting pump speed of response, we came to the following conclusions:

1. – The spring constant Ksp is defined by regulating accuracy requirements and stability constraints, result of a trade-off process to reach its final numerical value.
2. – Parameter Bvf which depends on pump physical parameters $Bvf = \mu l h / \delta$, can be controlled by designer, and can be modified by re-designing the pump.

We suggest the following re-design procedure:

1. – Substantially increase the clearance δ, and decrease ring friction surface h x l (4 π R l), obtaining a much, much smaller Bvf parameter.
2. – Eliminating the pressure drop Δp across the ring, thus eliminating (minimizing) leakage Qleak.
3. – Utilization of a piston to balance the pressure setting spring and doing so obtain higher flexibility in choosing the spring constant Ksp.

The following figure shows the new pump regulating ring, the balancing piston, and the two pressure equalizing chambers Pi and Po.

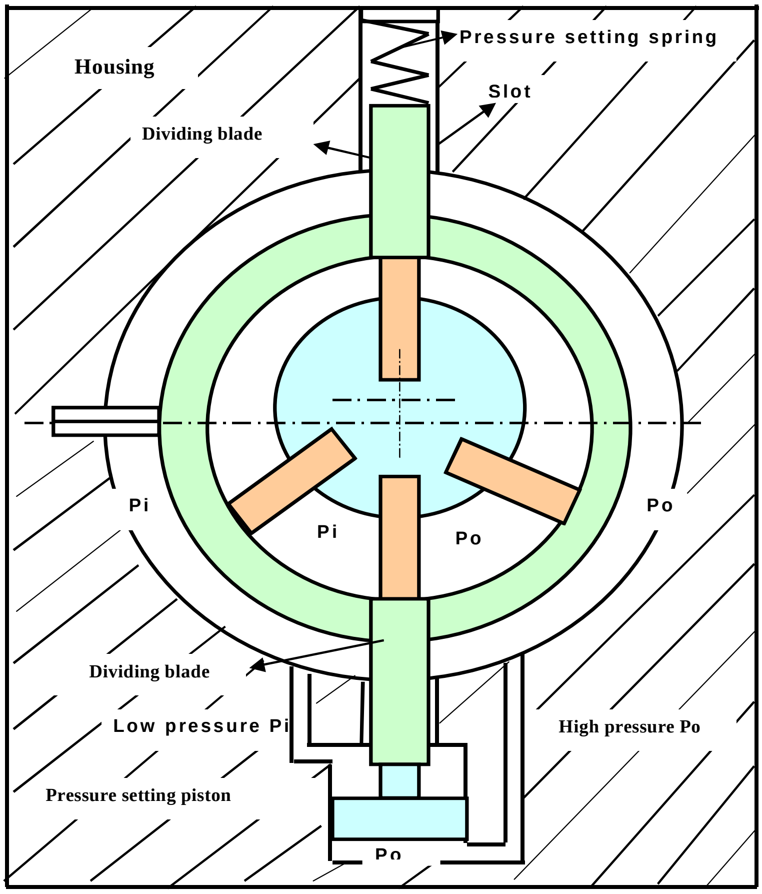

Figure 6: The "new pump" with the re-designed ring

The pressure equalizing chambers Pi and Po eliminate Qleak which is now limited across the two dividing blades, in the slots close to the spring and to the piston. Blades and ring are manufactured in one piece.

Rotor, vanes and dividing blades have the same width H and ring width is $h = H - 2\delta$. Clearance δ is now large, enables oil to enter both sides of the ring to be pressure balanced ($\Delta p = 0$ and Qleak = 0). Side force is balanced by the thrust screw, so dividing blades can move freely in their slots. Since the ring is pressure balanced, the regulating force is zero, vertical regulation is provided by the newly designed piston actuated by the differential pressure Po – Pi. Parameter Kpv is replaced now by the piston area Ap which can be designed at will.

Before we conclude the paper with a quantitative evaluation of an improved pump under the new conditions, let us summarize what are the achievements of our study and the concurrent re-design.

1 – Ring is pressure balanced $\Delta p = 0$, in both, high and low pressure chambers. The only forces on the ring are spring and piston, so it is designed with smaller thickness (5 mm not 9.5 mm) and smaller mass M.

2 – No case drain leakage Qleak ~ 0 (or close to zero). Almost no leakage means energy saving and overall higher pump efficiency. The leakage across the dividing blades is negligible, parameter K1k is negligible.

3 – Regulating method is now based on piston force Ap* (Po–Pi) balancing the spring force Fs + Fd. We can control regulating force by varying the area Ap, which enables enormous flexibility in choosing spring Ksp.

4 – Finally, we decrease substantially the friction area between the ring and the plates (h * l) at least by an order of magnitude, and control the gap δ, so the friction coefficient $Bvf = \mu\, l\, h / \delta$ is decreased significantly.

SUMMARY OF RESULTS

We shall summarize now the concluding results of our study / investigation relating to a specific pump with the following parameters, see figure 7.

- δ new = 2 x δ old
  or greater, depending on the oil viscosity
- Bvf new = 1 / 20 Bvf old
  after substituting the new parameters
- M new = 1 / 2.3 M old
  smaller ring mass due to new design
- Gss = - 1 / [K1k + N Ke (Kpv / Ksp)]
  Steady state gain which was dominated by the ratio Kpv / Ksp is now controlled by Ap / Ksp.
- Ksp old = 30 N / bar
  obtained from accuracy / stability considerations
- Ksp new = 1.75 Ksp old
- D = 25mm (A=490 mm^2) Piston of diameter (Area)

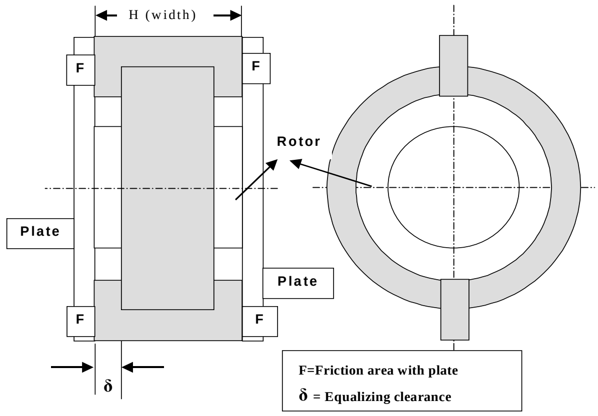

Figure 7: The new ring

The new design gives the designer an additional degree of freedom and grate flexibility to control pump speed of response without effecting accuracy and disturbance resistance. The final result of our study and re-design in Bvf / Ksp ratio is a tremendous improvement in pump dynamic response (band width) !

Bvf / Ksp new = 1/35 Bvf / Ksp old

CONCLUSIONS AND FURTHER STUDY

- The paper describes the application of control theory, analysis, evaluation and improvement of vane pump based on a Dynamic Model provided in the literature.
- The analysis concluded in the pump re-design, several of its important parameters, and in significant improved pump dynamic behavior (fast speed of response).
- In order to validate the results, a prototype has to be built and tested to study the actual pump behavior.
- The improved high bandwidth vane pump could now be utilized in high performance hydraulic servo systems.
- The new pump could be a part of the new generation hydraulics- Integrated Hydraulic System (pump, control servo-valve, actuator, sensors integrated in one unit).
- The governing mathematical model, the algorithms defining pump behavior, should be implemented in software and computer analyzed for optimization.

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