Optimization of an experimental hydraulic system with two flow control variables

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Abstract. Efficiency optimization is essential in many areas of engineering and system design. This work examines the efficiency of an experimental hydraulic system from different aspects: mechanical efficiency, pumps efficiency and flow stability. In this work, the flow control system depends on two independent variables with additional four degrees of freedom in terms of working pumping stations and flow path. An optimization algorithm was developed in order to identify the optimal duty point in terms of the efficiencies and stability within all possible system configurations. The results of the model are presented on several plots covering a 3D overview plot showing the system capabilities and additional 2D plots addressing each efficiency and stability optimization results and also demonstrating the system behaviour under different working conditions. The results of this work enables the user to easily choose the amount of working pumps and the duty point in order to achieve the desired flow rate. Moreover, it enables working in the desired optimal efficiency (or stability) in the proposed complicated two variable control system.

1. Introduction
Efficiency optimization process is a necessity in every industrial system design for various needs such as energy consumption, prolonging maintenance periods as exampled in [1-3] and many others.

In this work, an experimental hydraulic system optimization with two flow control parameters was accomplished. The system consists of three 4” parallel piping lines (#: 1,2,3). Each line contains a VFD (Variable Frequency Drive) driven centrifugal pump P# (each rated for 180 m³/h), which circulates water through a throttle control valve V#. The three lines are summed into an 8” header which delivers the flow through the load vessel (L1). Each line contains a vortex flow meter (FT#) and the total flow is defined as the sum of all three. An illustration of the hydraulic system is presented in figure 1.

![Hydraulic system scheme](image)

Figure 1. Hydraulic system scheme.
The system is designed to supply a wide range of flow rates to the load vessel, between 10-450 m³/h with an accuracy of ±2% (of the desired flow rate). Optimization calculations are based on the system's mathematical model which is thoroughly described in a previous study [4].

In order to achieve the wide desired flow range, the system may operate in four working modes (configurations): a) P1 line is working; b) P1 and P2 lines are working; c) P1, P2 and P3 lines are working and d) P1 is working and Bypass Valve (V11) is open. Since there are two degrees of freedom [denoted as (V,f)] in each working mode: opening percentage of the throttle control valve (V) and the electrical frequency of the pumps (f), there are several possibilities to achieve a certain flow rate. For example: for a single pump without bypass, flow rate of 160 m³/h can be achieved by ≈(80%, 38Hz), ≈(48%, 50Hz) and other duty points as well. Moreover, this flow rate may exist in other working modes. This variety of possibilities makes it difficult for the system operator to determine, for a certain flow rate, which configuration and duty point is the most suitable. Additionally, it is even more complicated to determine within these possible combination which is better. Hence, an optimization algorithm to determine the best working conditions for every desired flow rate (in all working modes) was developed.

After reviewing former studies, the applications where wide flow range is required are usually associated with chiller water supply as demonstrated in [5-6]. However, no similar application resembling the experimental systems discussed in this work was found. Hence, the method proposed here is genuine.

In this study, the optimization process was designed for three applicable parameters: system's mechanical efficiency, pump's mechanical efficiency; and since this system is an experimental system, a measure of the optimal flow stability was calculated as well. In order to perform the optimization process, an optimization module was added to the code of the basic mathematical model described in [4]. The code was written in Octave™. The output of the optimization process enables the user or the control system to choose the most fitting working mode and duty point according to the defined optimization.

2. Modeling methods

This section elucidates the methods and assumptions that served the optimization process. As mentioned in the previous section, the basic mathematical model is thoroughly described in [4] and won't be detailed in this work. The computing algorithm is divided into two sections: efficiency calculations and optimization process. The latter relies on the first. A flow chart, illustrating the calculation and optimization methods, is shown on figure 2.

2.1. Efficiency calculations

The algorithm starts with computation of the flow matrix Q(V,f). This matrix (sized NxM) represents all of the possible flow-rate combinations for a specific working mode. The value of flow is dependent on the values of two control variables (Duty Point): valves opening percentage (V) and pumps frequency (f). In the following phase, the efficiencies (Mechanical, Pumps) and the measure of flow stability are calculated at every duty point resulting in corresponding efficiencies matrices. The calculation is performed in the following manner:

2.1.1. Pump's efficiency

Pumps efficiency curves usually appears in the pump's manufacturer's datasheet. In this work, the nominal efficiency curve was fitted by a 2nd order polynomial equation for the pumps used in [4]. Adaptation of the efficiency curve for different working frequencies was accomplished by using the efficiency affinity law [7]:

\[
\eta_s = 1 - (1 - \eta_0) \left( \frac{f}{f_0} \right)^{0.1}
\] (1)
where $\eta_s$ is the calculated efficiency at frequency $f_s$ and $\eta_n$ is the nominal efficiency at nominal frequency $f_n$. In order to comply with the limitations of eq. (1), the simulated minimal working frequency was not lower than 50% of that of nominal frequency (50Hz).

Note that pumps efficiency is calculated for every duty point (Pumps's frequency and Valve’s opening percentage) in each working mode.

2.1.2. System efficiency

The efficiency of the system $\eta_{System}$ is defined as the ratio between the mechanical power delivered to the load vessel and the mechanical power supplied by the pumps:

$$\eta_{System} = \frac{P_{Load} \cdot Q_{Load}}{P_{System} \cdot Q_{System}} = \frac{k_{Load} \cdot Q_{Load}^3}{k_{System} \cdot Q_{System}^3}$$  \hspace{0.5cm} (2)

In eq. (2), $P$ represents the Pressure, and $k$ and $Q$ (detailed in [4]) are the mechanical resistance and flow rate of the load and the system respectively. Note that $Q_{Load}$ and $Q_{System}$ are not equal only when the bypass line is opened.

2.1.3. System stability

The system stability is defined as the absolute value of the divergence of each point in the flow matrix: A higher slope curve is less stable than a moderate one. The divergence is calculated using finite differences method (central difference):

$$|\Delta Q| = \sqrt{\left(\frac{\partial Q(V,f)}{\partial V}\right)^2 + \left(\frac{\partial Q(V,f)}{\partial f}\right)^2}$$  \hspace{0.5cm} (3)

The discrete divergence calculation was performed as follows:

$$\frac{\partial Q(V_n,f_m)}{\partial V} = \frac{Q(V_{n+1},f_m) - Q(V_{n-1},f_m)}{2\Delta V}$$
$$\frac{\partial Q(V_n,f_m)}{\partial f} = \frac{Q(V_n,f_{m+1}) - Q(V_n,f_{m-1})}{2\Delta f}$$  \hspace{0.5cm} (4)

The boundary condition is defined as the Dirichlet condition. Consecutively, the value of $|\Delta Q|$ is calculated with backward or forward difference according to the data in the matrix. An example of boundary conditions calculation with respect to the valve opening derivative is presented in Eq. (5). Calculation of the boundary for the frequency derivative is done similarly.

$$\frac{\partial Q(V_n,f_m)}{\partial V} \bigg|_{V_{n\pm1}} = \frac{Q(V_{n+1},f_m) - Q(V_{n-1},f_m)}{\Delta V}$$
$$\frac{\partial Q(V_n,f_m)}{\partial f} \bigg|_{V_{n\pm1}} = \frac{Q(V_n,f_{m+1}) - Q(V_n,f_{m-1})}{\Delta f}$$  \hspace{0.5cm} (5)

3. Optimization

Prior to performing the optimization process, an algorithm which finds the possible combinations of valve opening percentage and frequency is executed for every desired flow rate. The specific flow rate ($Q_k$) is within the desired range of [10,11,...,450] m$^3$/h (minimum to maximum). Mathematically, the algorithm finds the space of solutions within $Q(V,f)$ which satisfy the desired flow $Q_k \pm 0.5$ m$^3$/h (a delta is presented in order to prevent an empty solutions). The solution is given in two dimensional matrix $Q(V,f)$ which consists of two space vectors $V_k$ and $f_k$ representing the sets of valve opening percentage and pumps frequency which results in $Q_k \pm 0.5$ m$^3$/h.

Using the values of space vectors $V_k$ and $f_k$ within the calculated efficiency matrices will result in appropriate efficiency matrices corresponding to $Q_k$. The optimization process finds the "best" result within these matrices. For each working mode, the optimal system efficiency is defined as the
maximum of SystemEfficiency \((V_k', f_k')\), optimal pump efficiency as maximum of PumpsEfficiency \((V_k', f_k')\) and the most stable point is defined as the minimum of SystemStability \((V_k', f_k')\): minimal slope (minimal change in flow when a perturbation is introduced).

**Figure 2.** Flow-chart of efficiency calculation algorithm.

### 4. Results

Prior to representing the optimization algorithm results, we plotted a family of 2D curves representing the flow with respect to frequency and with respect to the valves opening percentage as seen in figure 3 for a single pump configuration. These plots can be used as a planning tool (regardless of the optimal desired state) to achieve a certain flow rate. Moreover, a look-up table can be easily implemented in the controller of the system.

**Figure 3.** 2D Flow curves for a single pump configuration. (a) – Flow Vs. Frequency for various valve opening percentage curves ; (b) Flow Vs. valve opening percentage for various Frequencies curves.

The optimization process resulted in 3 sets of two space vectors representing the optimal valve opening percentage and pumps frequency for each desired efficiency. For comparison purposes, all three optimization lines were plotted on the 3D duty curve (representing all the possible flow rates in
the system with respect to the valves opening percentage and the pump's frequency) as seen in figure 4 for a working mode of a single pump. Each line on the plot represents the optimal value of selected efficiency for a specific flow. Similar plots for all working modes can be generated as well.

**Figure 4.** Optimal Efficiency lines illustrated on a 3D plot of the flow for a single pump.

It is clearly seen from the figure that in order to achieve optimal mechanical efficiency, one must open the valves (and restrict the losses caused due to friction upon it). Similar action is required for pump's efficiency as well. However, in terms of better stability, the optimal duty points tend to higher frequencies and lower opening valve percentages.

**Figure 5.** Optimal: (a) Stability ; (b) Mechanical efficiency ; (c) Pumps efficiency for all working modes.

For ease of use from a user perspective, we decided to plot each of the efficiency lines for all of the possible working modes on one plot as seen on figure 5. Each of the plots in the figure represents the optimal value of desired efficiency for every flow rate within the range. This enables the user to easily choose the optimal working mode for the desired choice of efficiency/stability and the desired flow rate.

It can be seen from the figure that the system is more stable when less pumps are activated. A careful selection is required for overlapping low flow rates in different modes while examining the system's efficiency. In terms of pumps efficiency, higher efficiency is achieved for low flow rates when the bypass line is open.

**5. Conclusions**

In this work, we calculated the mechanical efficiency, pump efficiency and stability of a hydraulic system with two control parameters (Pumps frequency and throttle valves openings percentage) and four degrees of freedom in terms of number of active pump stations and an introduction of a hydraulic bypass line. Consecutively, we calculated the optimal duty point in terms of the calculated efficiencies.
mentioned above for all possible working conditions considering the all the degrees of freedom. The output of the optimization process enables the user to choose the most efficient duty point in terms of the desired optimum. Alternatively, these results can be easily implemented into the controller of the system in the form of look-up tables to allow for automatic navigation to the desired flow rate within the desired optimum.

Although this work is strictly theoretic, it is based on well-known hydraulic equations and theory and hence the presented results are satisfying. Nevertheless, we intend to perform experiments in the future to further validate the model results.

In summary, this work introduced an algorithm which optimizes several parameters using analytical equations and showed good results. This method may be easily applied to other systems during the design and optimization process.

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