Innovative fan drive solution for mobile machines

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Abstract

Hydraulic fan drive systems offer advantages for thermal management in mobile machinery, particularly in construction and agricultural equipment applications. Compared to traditional mechanical or direct-drive electric systems, hydraulically actuated fans enable flexible installation, provide good control and reduced power consumption by decoupling fan speed from engine speed. In the design of hydraulic fan drive systems for mobile machinery, optimizing the design to be both compact and affordable is essential for meeting customer expectations. For the status quo, two independent valves are required to achieve both fan speed control and radiator cleaning functionality provided by fan reversal. Simplifying the hydraulic system by using one valve instead of two would enable for size and cost reduction. This paper presents a patented and innovative hydraulic fan drive solution incorporating a single integrated valve in an external gear motor to control both fan speed and reversing direction. The objective of this paper is to verify the feasibility of the one-valve solution by means of a 1D simulation model. Simulation results demonstrated that the one-valve solution, when integrated with its enhanced feed-forward control, delivers performance comparable to the current two-valve solution, particularly in dynamic response and maintaining consistent flow characteristics despite pump speed fluctuations. This provides compelling evidence that the one-valve solution offers a simplified, cost-effective, and dynamically robust alternative for future mobile hydraulic systems.

Keywords. Mobile hydraulics, Fan drive.

1. INTRODUCTION

Mobile machinery applications, such as construction and agricultural equipment, operate in competitive markets where cost efficiency and system compactness are critical factors in equipment design. These machines generate heat during operation, primarily from the engine and the hydraulic system itself. Thermal management is necessary for maintaining optimal operating temperatures and it is achieved by matching heat rejection with minimal energy consumption. The engine's heat is absorbed by coolant fluid, then carried to the radiator, where the heat is exchanged with ambient air [1]. For diesel engines of mobile machinery, it is estimated that the power dissipated in the heat exchanger is 45% of the power output [2] and 30% of fuel energy [3]. To improve heat rejection, a cooling fan actively forces the air across the radiator. The faster it spins, the more air it pulls through the radiator, and the more engine power it consumes. Therefore, regulating the fan speed is required for efficient thermal management.

Fan drives can be mechanical, electric, or hydraulic. Mechanical systems, using pulleys and belts, often overcool due to fixed speed ratios. Electric drives suit low to medium power demands. Hydraulic fan drives are a common solution for heavy-duty off-road vehicles due to their power delivery, ability to control fan speed independent of engine speed and assembly flexibility [1]. This ability means that the fan is not wasting energy by rotating faster than it needs to, just because the engine is running at high speed. This work focuses on hydraulic fan drives.

Optimizing the design of hydraulic systems used in fan drive application to be compact, and affordable is essential for meeting customer expectations. It needs to be compact to fit in increasingly physically optimized machines and affordable to keep up to the market competitiveness [4]. However, the design should still meet functional and reliability performance. As one of the world's leading suppliers for drive and control technologies, Bosch Rexroth offers, among other things, hydraulics, electric drive and control technology, gear technology as well as linear motion and assembly technology. In the fan drive application, the company established a strong presence with its solutions featuring external gear motors and valves manifold. To maintain competitiveness, two strategic approaches were considered for product improvement.

As presented in Figure 1.1, the architecture of the hydraulic system for fan drive is defined by three subsystems: 1) hydraulic pump, usually powered by diesel engine, 2) hydraulic motor, which drives the fan, and 3) valves, when specific functions are necessary, which can be in an external manifold or integrated in the motor. Integrated options are increasingly used in the market because they optimize space, reduce the number of pipes and hoses, and decrease number of connections, which are potential leak points.

Different combinations of hydraulic pumps and motors are possible. In this paper, the focus is a system with external gear pump with external gear motor, both with fixed displacements. For this combination, several possible functions can be implemented by valves such as:

- fan drive speed control,
- radiator cleaning functionality provided by fan reversal,
- cavitation mitigation for the period when the pump stops, and the motor continues to rotate for a certain time due to the fan rotational inertia.

The need for each of the functions depends on the specification of the machine and its application.

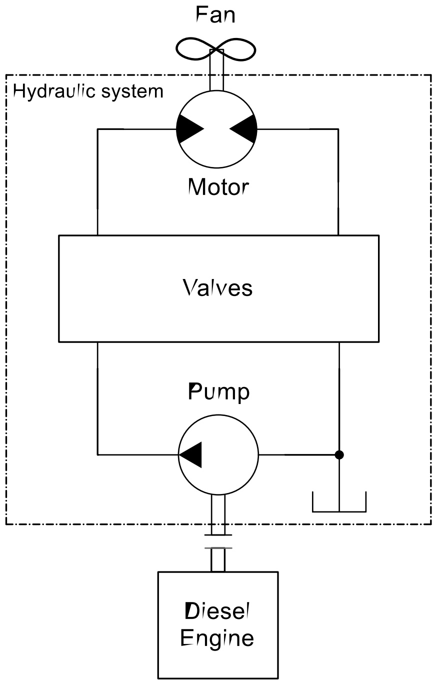


Figure 1.1 – Schematics of a generic hydraulic system for fan drive with fixed displacement pump and motor

In the status quo solution, available in the market, the speed control and the fan reversal is performed by two different hydraulic valves. This paper presents and patented and innovative solution that combines the two functions in one single valve proportional direction valve [5]. The novel concept is called then one-valve design and the status quo, two-valve design. Combining two functions in one valve allows for size and cost optimization.

The proportional directional valve is however sensible to input flow rate from the pump, which is a challenge. In mobile machinery the pumps are usually powered by diesel engines. If the diesel engine speed varies, the pump speed also changes and consequently the input flow rate. For the novel solution, this could cause then a variation in the fan speed, which is not desirable. To overcome this, a feedforward controller is proposed.

Although, the motivation behind the one-valve design is to reduce size and cost, in this paper the objective is to verify the feasibility, by means of 1D simulation, that the novel one-valve design with feedforward controller can have similar or better functional performance than two-valve design, which is the current status quo. Representative operation cycles derived from typical mobile applications will be defined and applied to the model to compare its performance to the two-valve solution.

In the following next section, it will be explained how the status quo design works. Later, a novel concept will be explored the by explaining the difference to status quo, the challenged raised by it, and how to overcome it. Then, the simulation model developed, and the simulation results for the performance verification will be presented and discussed. Finally, the main findings and outline the next steps will be summarized.

2. Status-quo Solution

A hydraulic system for fan drive may be required to perform several distinct functions to meet the full range of operational demands. The specific system utilized for this investigation and presented in Figure 2.1 is designed with three primary capabilities delivered by valves: fan speed control for thermal management optimization, a reversing function for radiator cleaning, and an integrated anti-cavitation to protect the hydraulic motor.

In the status quo solution, the speed control is performed by proportional pressure relief valve (V1) and the fan reversal by on/off 4/2 directional valve (V2). The status quo solution using two independent valves (V1 and V2) inherently increases system size, and cost. Typically, an anti-cavitation valve is used for cavitation mitigation. In this paper no change is proposed for this function.

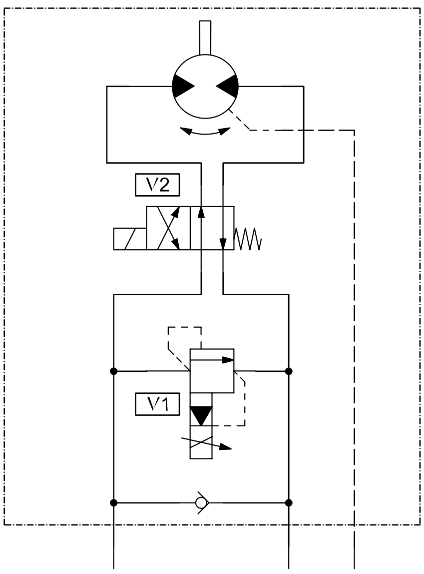


Figure 2.1. Hydraulic schematics of the status quo two-valve design (pump is omitted)

2.1. Speed control

Thermal management, which is crucial for maintaining optimal operating temperatures in mobile machinery, relies on cooling fans actively forcing air across the radiator. The rate of heat rejection is directly proportional to the volumetric airflow through the radiator, which in turn is directly proportional to the fan rotational speed. In pursuit of higher operational efficiency, the control system is designed to regulate the fan's rotational speed to the minimum level required by the current thermal conditions [1]. The logic of speed control using a proportional pressure relief valve is shown in Figure 2.2.

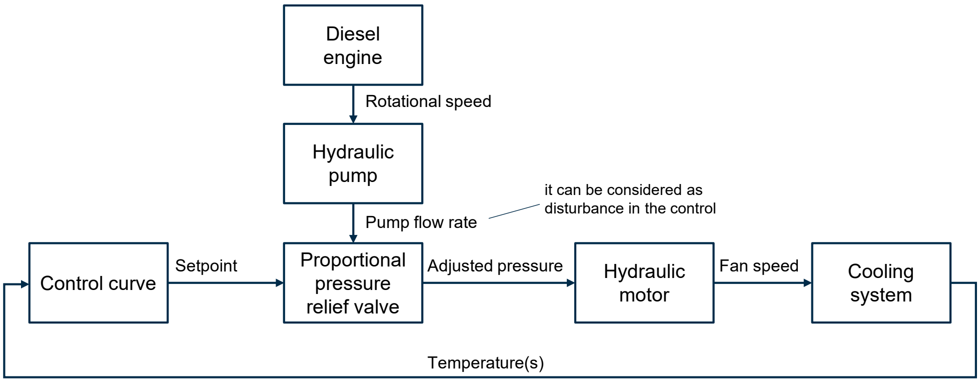


Figure 2.2. Speed control logic using proportional pressure relief valve

At lower temperatures, a reduced cooling airflow is needed, and therefore a lower fan speed is sufficient. The relationship between system temperature and required fan speed is typically defined by a control curve (often referred to as a "cooling curve") defined to meet machine's specific cooling requirements. Figure 2.3 illustrates an example of cooling curve. Fan can be set to standstill if the temperature is below a defined threshold [6].

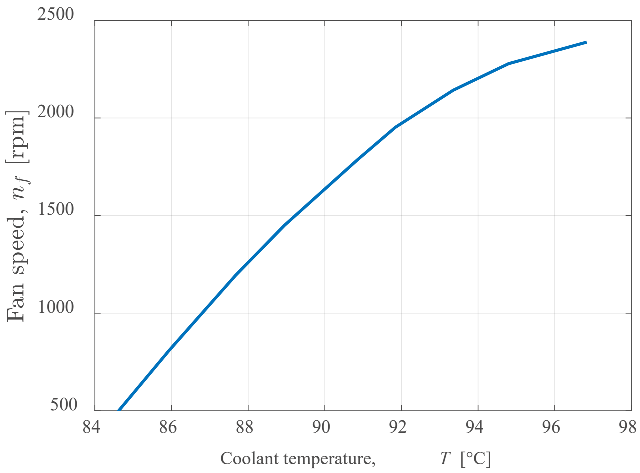


Figure 2.3. Typical fan speed control curve as a function of engine coolant temperature, adapted from [3]

The proportional pressure relief valve (V1 of Figure 2.1), which modulates system pressure in response to an electrical setpoint. Because the motor's torque is a direct function of the pressure differential across it, controlling the pressure effectively controls the torque. Since primary load on the motor is the fan's aerodynamic drag, it requires greater torque to achieve higher speeds. Consequently, a direct relationship is established where system pressure, as regulated by the valve, dictates the fan's rotational speed.

To ensure the engine does not overheat during an electrical failure, the system is designed to be fail-safe. If the control signal is lost, the fan automatically runs at maximum speed. This is achieved using negative control logic: as the electrical signal increases, the fan speed decreases. Therefore, a zero signal commands maximum fan performance. As presented in Figure 2.4, the Bosch Rexroth KBVS.3B valve is a common example of a component that uses this principle in fan drive systems.

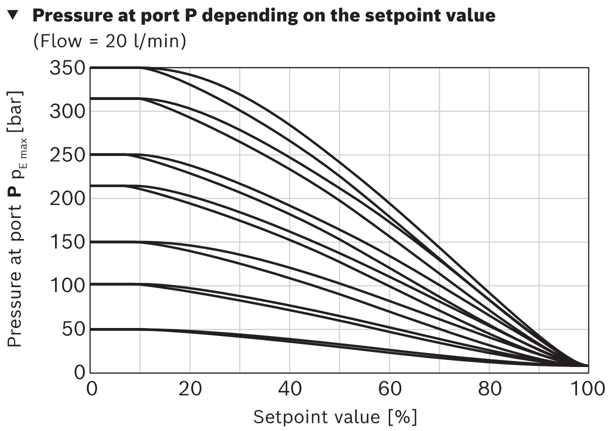


Figure 2.4. Pressure to setpoint relationship of KBVS.3B [7]

While a constant electrical setpoint to the proportional pressure relief valve should ideally maintain a constant system pressure and, therefore, a stable fan speed, the valve's inherent pressure-flow (P-Q) characteristic (Figure 2.5) causes a deviation from this ideal behaviour. As the pump's output flow increases – typically due to an acceleration of the diesel engine – a greater portion of this flow must be diverted through the valve. According to the P-Q curve, this results in in a change in system pressure even though the electrical setpoint remains fixed. This effect makes the fan speed to change under variations in pump speed. This inherent sensitivity of the status quo design will serve as the performance benchmark against the novel solution.

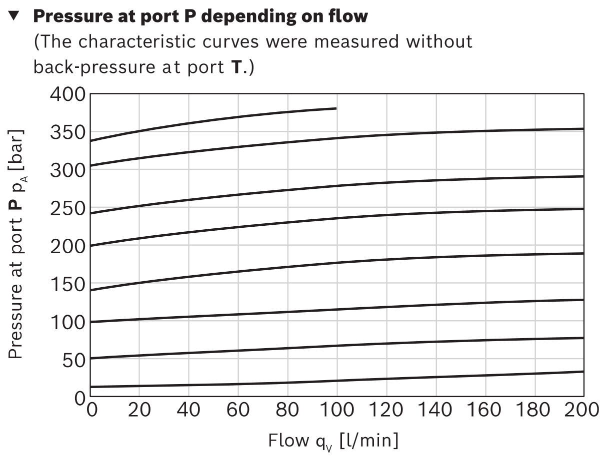


Figure 2.5. Pressure to flow relationship of KBVS.3B [7]

2.2. Reversal of speed

In many off-highway and agricultural applications, machinery operates in environments with high concentrations of airborne particulates, such as dust, chaff, and other debris. This debris can accumulate on the surface of the radiator, clogging the cooling and severely impeding airflow. A clogged radiator leads to a reduction in heat exchange efficiency, which can cause the engine and other hydraulic systems to overheat, reducing overall productivity and potentially causing damage to critical components [1].

To mitigate this issue, different radiator cleaning solutions have been developed, mainly by forcing air in the opposite direction to dislodge accumulated debris and carry the dirt away. Two usual methods are 1) changing the pitch of the fan blades [8] and 2) reversing the fan's direction of rotation [1]. This paper will focus on the latter approach, where the fan's rotational direction is reversed to generate a burst of air in the opposite direction of normal operation.

In the hydraulic system under investigation, this reversing function is accomplished by changing the fluid path to the hydraulic fan motor. As presented in Figure 2.6, this is typically achieved using a 4/2 on/off directional control valve that switches the pressure and return lines. During normal cooling operation, the pump's pressure port P is connected to working port A of the motor, while the port B is connected to the reservoir T. To initiate the cleaning cycle, the solenoid “a” is actuated, valve shifts, connecting the pressure port P to the working port B and working port A to reservoir T. This change in flow direction causes the motor and, consequently, the fan to rotate in the opposite direction.

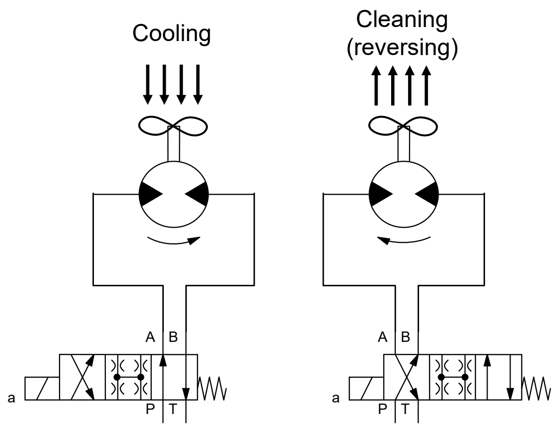


Figure 2.6. Directional positions for cooling and cleaning (reversing) modes

The activation of the reversing cycle is managed by the machine's controller. This can be initiated manually by the operator via a button in the cab or programmed to occur automatically at periodic intervals. The frequency of these automatic cleaning cycles can be tailored to the specific application and expected level of debris.

3. NOVEL SOLUTION

The idea behind the novel solution is to use a simplified hydraulic system combined with a feedforward control system to achieve the functionalities required for the fan drive application with similar performance as status quo solution. As shown in the hydraulic schematics of Figure 3.1, in the novel concept, the speed control and fan reversal functions are combined in one single proportional directional valve (V3). Therefore, the novel concept is called one-valve design. The spool design is explained in section 3.1 and the feedforward controller in section 3.2.

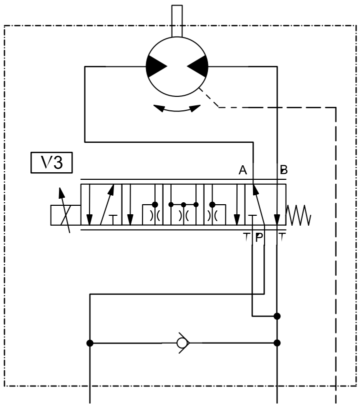


Figure 3.1. Hydraulic schematics of the novel concept one-valve design (pump is omitted)

3.1. Spool valve design

This novel solution employs a single proportional directional valve whose spool geometry is designed to achieve both fan speed control and direction change. This valve features a spool specifically designed for fan drive applications through control edges to achieve the performance required. The spool position can be continuously set from 0 to 100% by an electrical signal, which is referred as setpoint. Figure 3.2 illustrates a simplified representation of the spool valve geometry and its 5 functional stages.

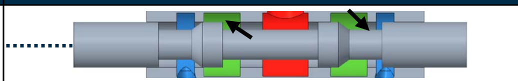
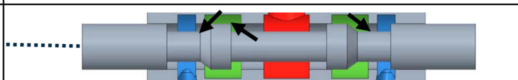
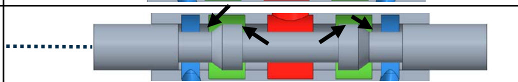
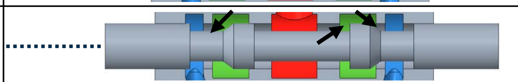
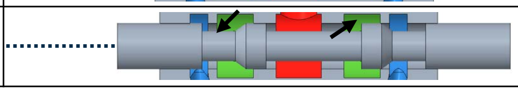
Stage	Setpoint	T A P B T	Mode	Fan speed
1	0%		Cooling	Maximum
2	25%		Cooling	Intermediate
3	50%		Standstill	Stop
4	75%		Cleaning	Intermediate
5	100%		Cleaning	Maximum

Figure 3.2: Simplified representation of the spool in its five stages

Please find below an explanation of the five stages:

1. To meet the fail-safe requirement, at “zero” setpoint the maximum fan speed in the cooling mode is achieved by connecting P to A and B to T. The meter-in (P to A) edge is designed to achieve pressure drop of ~3 bar at 40 L/min to meets the market requirements, which is similar to other parts in body solutions of Bosch Rexroth such as SM12 open control block [10]
2. By increasing the setpoint, the P-A and B-T connections are maintained, but the A-T connection partially starts to open, causing some of the flow that would go to the motor to now divert to the reservoir. This causes the motor to decrease its speed, but still kept in the same direction and therefore cooling mode.
3. When the setpoint reachs 50%, then all four ports PABT are connected by specific designed metered edges. This allows for a smooth transition when reversing. Also, if the controller keeps the 50% setpoint, then the motor is set to standstill mode, i.e., it stops.
4. By this stage, the reversing starts connecting P to B and A to T, but not at full speed since there is a connection between B to T, as a mirror of stage 2. Therefore, this stage is in the cleaning mode with partial speed.
5. When the connection between B to T closes, then the fan operates at cleaning mode at full speed, as a mirror of stage 1.

To control the speed of the fan, the flow rate going to the motor (Q_m) is reduced by diverting part of the pump flow rate (Q_p) to reservoir, which is known as bleed-off flow rate (Q_b) as shown in Equation 3.1.

$$Q_m = Q_p - Q_b \qquad (3.1)$$

This technique allows for flow control without throttling, i.e., restricting, the path from pump to the load [11]. The challenge of the proposed flow control is the hight sensitivity of the controlled motor flow rate to the pump flow rate. To explain, if the speed of the diesel engine increases, the pump flow rate also increases, and if the bleed-off flow rate remains the same, the motor flow rate and consequently the fan speed will increase, which is not desired. As will explained in the next section, to overcome this sensitivity, a feedforward controller is developed where it takes account the diesel engine speed as input to keep a stable fan drive speed.

3.2. Control with feedforward

The objective of the control with feedforward is to keep the fan speed stable, even with fluctuations in pump speed. As shown in the Figure 3.3, it calculates the spool position setpoint (x') based on the fan speed setpoint (n_f'), determined from control curve according to the measured temperature (T), and adjusted to the measured pump speed (n_p), which is commonly available in the electronics of modern machines. It allows to the controller to automatically adjust the bleed-off flow rate to compensate for a possible fluctuation in pump flow rate (Q_m), which keeps a constant controlled flow rate to the motor (Q_m), therefore a stable fan speed (n_f).

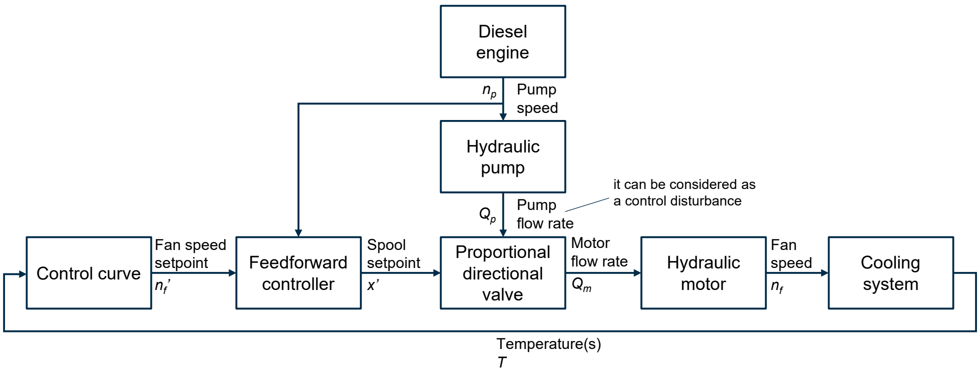


Figure 3.3. Speed control logic with proportional directional valve

To understand behind how the control calculates the spool setpoint based on the fan speed setpoint and the pump speed, the Equation 3.1 is reorganized to isolate the bleed-off flow rate (Q_b):

$$Q_b = Q_p - Q_m \tag{3.2}$$

Knowing the measured pump speed (n_p), the pump displacement (V_p), and the volumetric efficiency of the pump ($\eta_{p,v}$), it is possible to calculate the flow rate as follows:

$$Q_p = n_p V_p \eta_{p,v} \tag{3.3}$$

Similar way, it is possible to calculate the motor flow rate (Q_m) knowing the desired motor speed, which is the same as the fan speed setpoint (n_f'), the motor displacement (V_m), and the volumetric efficiency of the pump ($\eta_{m,v}$):

$$Q_m = n_f' V_m \eta_{m,v} \tag{3.4}$$

Consolidating Equations 3.3 and 3.4 in 3.2:

$$Q_b = n_p V_p \eta_{p,v} - n_f' V_m \eta_{m,v} \tag{3.5}$$

The bleed-off flow rate is controlled by the spool opening, which can be described as a turbulent orifice [12]:

$$Q_b = C_d A \sqrt{\frac{2}{\rho} \Delta p} \tag{3.6}$$

Where C_d is discharge coefficient, normally considered 0,7, A the orifice area of the bleed-off control edge, which varies according to spool position, and (Δp) the pressure differential across the bleed-off control edge.

The pressure differential across the motor can be considered the same as the differential across the bleed-off control edge, since its orifice area is significantly more restrictive than the meter-in and meter-out. As explain in the section 2.1 for the status quo design, the relationship between pressure differential across the motor (Δp) and fan speed setpoint (n_f') is known:

$$\Delta p = f(n_f') \tag{3.7}$$

The orifice area of the bleed-off control edge (A) depends on the spool setpoint (x') and its relationship is defined in the spool design:

$$x' = g(A) \tag{3.8}$$

Combining Equations 3.5, 3.6, 3.7 and 3.8, then:

$$x' = g\left(\frac{n_p V_p \eta_{p,v} - n'_f V_m \eta_{m,v}}{C_d \sqrt{\frac{2}{\rho} f(n'_f)}}\right) \tag{3.9}$$

Equation 3.9 therefore governs the behaviour of the feedforward controller by continuously calculating the spool setpoint (x') according to fan speed setpoint (n'_f), and the measured pump speed (n_p).

4. SIMULATION MODEL

Having clarified how the status quo solution works, which is the performance benchmark, and the challenges involved in the novel solution, a simulation model was created to support the product development. The use of a simulation model significantly accelerates the design cycle and reduces costs by enabling early performance validation and optimization of spool geometry reducing the design iterations. The objective of the model in this phase is to confirm the feasibility of the novel solution by performing a preliminary assessment of system performance in comparison to status quo.

The model was created using AMESim, a mechatronic systems simulation platform [13]. For this comparative analysis, models for both the existing and novel solution were created. The schematic diagrams for both are presented in Figure 4.1 side-by-side, with key components labelled for clarity. Parameters used in the model such as the size of the pump, motor and fan are not from any specific machine, but those fall within the range commonly found in fan drive application. If necessary, in the future it is possible to parameterize the model to validate a specific configuration.

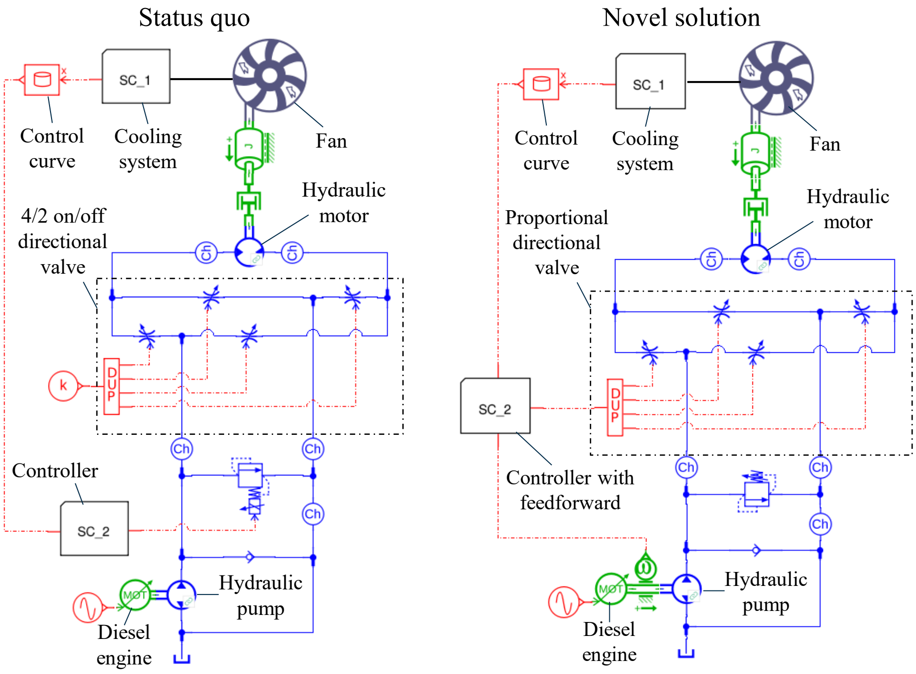


Figure 4.1. Schematics of simulation models

For the 4/2 on/off directional valve (status quo model) four orifices with turbulent conditions were used, one for each connection P-A, P-B, A-T and B-T. The orifice area (A) of each connection varies according to the spool setpoint (x'), where the relationship is defined by the spool geometry including the notches. The time response of the valve is defined by a first order response, where a 99% of the setpoint is achieve within 100 ms. Same approach is used for the proportional directional valve (novel solution model). It was considered a fan with a torque of 50 Nm at 2500 rpm. For the pump and motor, it was chosen Rexroth external gear units of 28 ccm and 19 ccm respectively, where volumetric efficiencies were taken from datasheet curves [14] [15]. The oil properties followed the ISO VG 46 classification. The influence of temperature variation on the parameters of the hydraulic components was not considered. Other models are standard from AMESim library. The main parameters of the model are available in Table 4.1.

Table 4.1. Main parameters of simulation model

Component	Description	Value	Unit
Motor	Displacement	19	ccm
	Volumetric efficiency	0.92	-
Pump	Displacement	28	ccm
	Volumetric efficiency	0.95	-
Fan Drive	Static friction	5	Nm
	Running friction	4	Nm
	Viscous friction coefficient	0.0184	Nm/rpm
	Moment of Inertia	0.07	kg.m²
Valves	First order time constant	0.0217	s
Proportional Pressure Relief Valve	Flow Rate Pressure Gradient	6.15	L/min/bar
Fluid	Fluid density	865	kg/m³

5. RESULTS

The performance of the novel solution (one-valve design) was evaluated under a significant disturbance by varying the pump speed and compared to the status quo (two-valve design), as illustrated in the Figure 5.1. To show that feedforward controller is indeed necessary, the novel solution behaviour was presented both with and without this controller. As presented in Graph b, a periodic disturbance was introduced by varying the pump speed sinusoidally between approximately 1,000 rpm and 2,400 rpm over a 20-second cycle. The fan speed of status quo, novel solution without feedforward, novel solution with feedforward is presented in Graph a. The objective was to maintain a constant fan speed of 1,400 rpm.

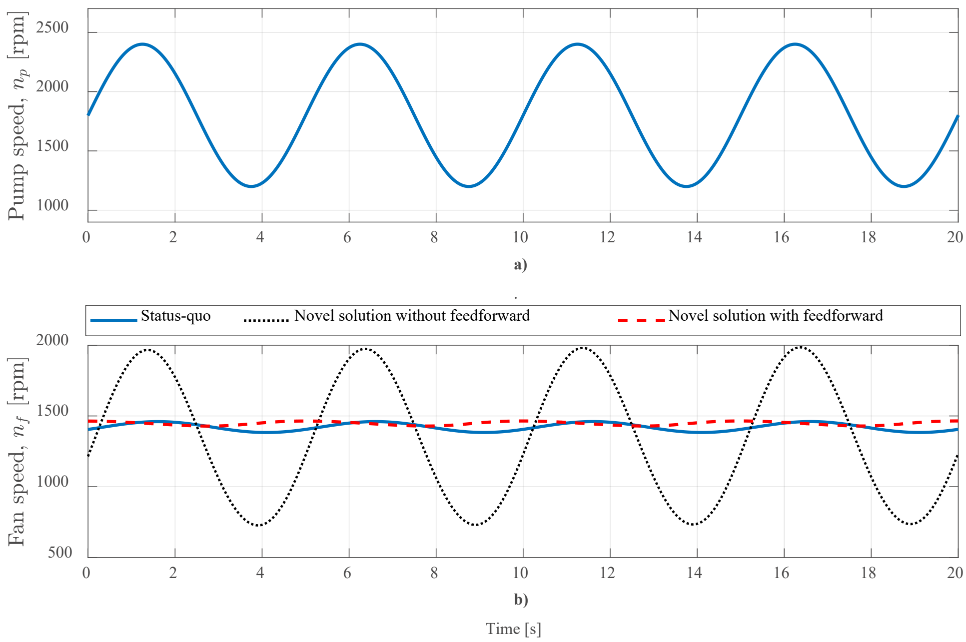


Figure 5.1. Fluctuation in fan drive speed due to variation in pump speed (simulation)

- **Status quo (benchmark):** The existing control system (solid blue line) serves as the reference for performance. It demonstrates partial disturbance rejection, with oscillations of approximately ±38 rpm amplitude.
- **Novel solution without feedforward:** When compared to the benchmark, this control strategy (dotted black line) was unable to compensate for the disturbance. The fan speed exhibited large, unstable oscillations with amplitude of approximately ±623 rpm, demonstrating a clear failure to regulate the system and a significant degradation in performance.
- **Novel solution with feedforward:** The proposed controller incorporating a feed-forward mechanism (dashed red line) demonstrated better performance than Status quo. It effectively attenuated the disturbance from the pump with oscillations of approximately ±18 rpm amplitude.

These results indicate that the inclusion of a feedforward control loop provides a substantial improvement in disturbance rejection and system stability over both the status-quo and non-feed-forward solutions. Other investigations were also carried out using the simulation model, such as the assessment of peak pressure and velocity during reversal, where the requirements were met However, these were not presented in detail since the focus of this paper is on the stability in speed control.

6. CONCLUSION

This paper successfully demonstrated by simulation, that the novel one-valve solution, when integrated with its enhanced feed-forward control, can overcome the challenge of fan speed instability caused by engine speed fluctuations when using proportional directional valves. The feedforward control strategy shows to be capable of providing robust disturbance rejection to pump speed variation, using the status quo solution, which is a pressure-controlled fan drive, as a performance benchmark.

Simulation results revealed that the status quo exhibited fan speed oscillations of approximately ±38 rpm under variable pump speeds. The proposed novel solution with feedforward controller effectively compensated for these disturbances by achieving ±18 rpm oscillation, while without controller it presented low performance with ±623 rpm oscillations.

The findings show that the use of a feedforward controller can enable the use of a cheaper and more compact hydraulic system, while maintaining low sensitivity to variations in engine speed. Furthermore, the development process underscored the value of simulation as a powerful tool for rapidly designing and validating control concepts prior to hardware implementation.

Future work should prioritize the experimental validation of these simulation results on a physical test bench to confirm the real-world performance of the hydraulic system and controller. Further investigations could also explore the system's robustness against other operational variables, such as changes in valve’s time response and geometry tolerances.

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