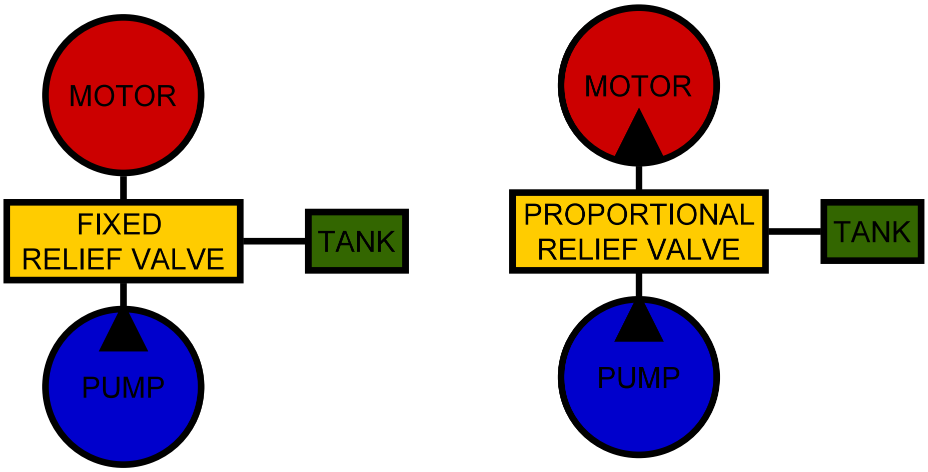

A typical configuration for a hydraulic fan drive system with fixed displacement pumps and proportional relief valves.

HYDRAULIC FAN DRIVES: IT'S NOT JUST ABOUT THE FUEL

BY CHARLES THROCKMORTON

As engineering students, we were introduced to the concept of efficiency in our first physics courses. Essentially, every process is inefficient — you have to put more energy or power into the process than you can hope to get out. Energy costs a lot, but it's more than that. We need to increase productivity using less power.

The mobile off-highway equipment industry is faced with two daunting tasks. One is to rise out of an economic recession, the other to simultaneously prepare to meet the requirements of new emissions regulations. Taken individually, each presents a challenge. Taken together, they're intimidating.

By reducing wasted energy or the parasitic losses in a system, more of the engine's power is available to produce useful work. When a cooling system is designed, it is sized to provide enough cooling to ensure that it can transfer the maximum heat rejection from the machine to the environment at the maximum design ambient temperature. However, very few machines will be applied in conditions that include maximum heat rejection and maximum ambient temperature simultaneously.

Since 1983, engineers at Sauer-Danfoss have been involved in applying proportional hydraulic fan drive solutions into both on- and off-highway applications. These early systems relied on thermostatic valves to regulate fan speed as a function of coolant temperature. Later — as machines became more complicated and more sub-system temperatures needed to be controlled — multiple electronic sensors, electronic controllers and proportional pressure control valves were added. Currently, fan drive controls are being integrated with engine and vehicle management controllers.

These proportional hydraulic fan drives have relied on regulating the pressure applied across the fan

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Fan Laws

$$\frac{FP_1}{FP_0} = \frac{Q_1 v_1 \Delta P_1}{Q_0 v_0 \Delta P_0} = \frac{n_1{}^3 D_1{}^5 v_1}{n_0{}^3 D_0{}^5 v_0}$$

Where:

$FP_{o,1}$ — fan power (HP, kW)

$Q_{o,1}$ — air flow rate (ft^3/min, m^3/sec)

$V_{o,1}$ — specific weight of air (lbf/ft^3, kg/m^3)

$\Delta P_{o,1}$ — pressure drop across the fan (in. Hg, in. H_2O, Pa)

$n_{o,1}$ — fan speed (RPM)

$D_{o,1}$ — Fan Diameter (ft, m)

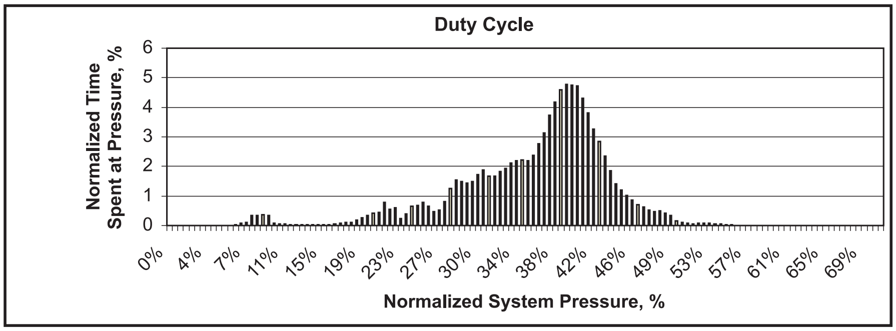

Fig. 1. A typical measured duty cycle for a machine operating at nominal summer ambient conditions.

drive motor to control fan speed. For any given combination of hydraulic motor and fan, there is a unique quadratic relationship between hydraulic pressure and fan speed, as defined by a family of equations known as the "fan laws."

Every machine is different, but it is not unusual for a typical machine to operate where the average load on the cooling system would require the proportional hydraulic fan drive to operate 80% of the time at less than 25% of the maximum design fan power (or 40% of the maximum design pressure). A typical measured duty cycle for a machine operating at nominal summer ambient conditions is shown above. For this test, the pressure drop across the motor was recorded; from this data, the histogram of Fig. 1 was produced.

Hydraulic pressure is much easier to regulate than flow rate. Over time, systems can degrade due to the effects of temperature and contamination. Degradation results in increased flow losses, or leakage, of the hydraulic components. If the control system were to regulate flow to the fan motor, any increase in leakage would cause the fan speed to fall below the desired value. By regulating pressure, the effects of contamination and/or viscosity are minimized. As long as there is sufficient flow to the hydraulic motor, the fan speed can be controlled by regulating the pressure drop across the motor. Consequently, the control limits that were originally installed into the controller will remain valid over the entire life of the machine and do not need to be reset (calibrated) in the event that a component is replaced at a service interval.

Hydraulic fan drives with fixed displacement pumps and proportional relief valves are the most common hydraulic fan drives in use today. These systems utilize fixed displacement pumps, fixed displacement motors and proportional relief valves to regulate the speed of the fan and control temperatures for engine coolant, hydraulic oil and charge-air coolers. The pressure setting of the proportional valve is regulated by controlling the current applied to a valves' integral solenoid by the fan drive controller.

The controller receives signals from temperature sensors, and/or switches, on the vehicle. These control systems may be simple or complex and are limited only by the imagination of the system designer. The system designer inputs temperature limits for each of the sub-systems into the controller. The controller reads the temperature sensors and sends a current to the solenoid to adjust the fan speed to maintain the temperature of all of the sub-systems within the design limits.

This version of the proportional hydraulic fan drive system can save a significant amount of power over the basic belt-driven fan by only turning the fan at the speed necessary to

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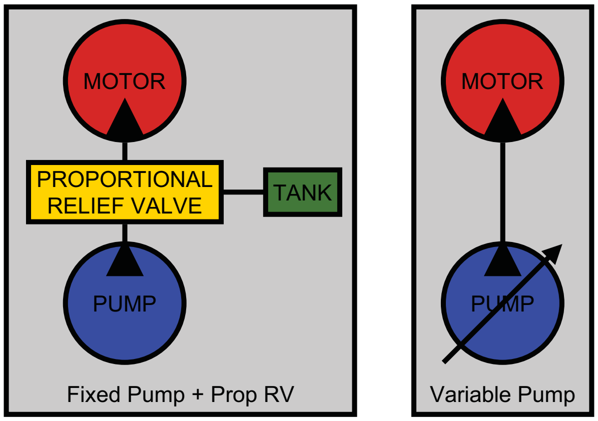

Hydraulic fan drive designs with variable displacement fan drive pumps and fixed displacement motors.

maintain the system temperature within the design limits. As the heat rejected by each system changes or as the local ambient temperature changes, the fan speed changes to maintain the coolant temperatures.

Hydraulic fan drives with tandem fixed displacement pumps are slightly more complex than the simple proportional system, but offer substantial power savings to the machine's system designer. In practice, pump No. 1 is sized to provide approximately 70% of the total displacement required to provide flow to the motor, and pump No. 2, the remainder. The fan drive controller determines the fan speed necessary to provide cooling for the system. When the controller recognizes that the commanded fan speed exceeds approximately 65% of the fan trim speed, it signals the dump valve to close. Before the dump valve closes, the flow leaving pump No. 2 is routed directly back to the pump inlet (or the reservoir) at minimum bypass pressure. This minimizes the flow in the hydraulic lines between the pump and the reservoir return port and the bypass flow across the proportional relief valve while the system requires minimum cooling. This will save even more of the engine power that would otherwise be wasted and converted into heating the oil in the reservoir.

Hydraulic fan drives with variable displacement fan drive pumps and fixed displacement motors are perhaps the most elegant solution for vehicle cooling systems. These pumps, specifically designed for the fan drive application, integrate open circuit technology with an integral proportional pressure control valve.

The proportional valve receives an electrical signal directly from the system's temperature controller. The outlet pressure of the pump is directly proportional to the command signal and the fan speed is regulated to provide the necessary cooling. This minimizes wasted power by delivering only the flow to the motor that is needed to rotate the fan. There is no bypass flow and passage losses are minimized.

For the example shown in Figure 1, hydraulic fan drives were sized to satisfy the maximum heat rejection requirements of a vehicle at maximum ambient conditions. This vehicle was operated at maximum engine speed to perform a normal function on a typical summer day in the United States.

For maximum cooling, the system would need to operate at 100% of the maximum trim pressure to provide cooling for maximum heat rejection at maximum ambient temperature. The trim pressure is the fan systems' maximum operating pressure. As mentioned previously, the average required fan power is determined by the vehicle's actual duty cycle and the actual ambient temperature. For an engine operating continuously at maximum speed, the average power of the belt-driven fan is equal to the maximum fan power.

The power to drive a belt-driven fan is dependent only on the engine speed. To illustrate the benefits of proportional hydraulic fan drives, in Table 1, we have compared a belt-driven fan with all the hydraulic fan drives mentioned earlier.

Of course, the system designer must consider the cost of the components in the system as well as the available space to install them. Compared to the fuel savings shown in the table, there are additional benefits for incorporating proportional hydraulic fan drive solutions. For example, the purchase cost of fuel is variable and is likely to increase over the life of the machine, but the real cost includes the transportation and labor to get the fuel into the machine at the job site.

The engine power that is saved can be utilized by the vehicle owner to increase their productivity as well as reduce the total cost of ownership. By minimizing the wasted power that goes to heating the hydraulic oil, the total heat rejection from the system

	Typical Fan Operating Duty Cycle:				
To satisfy the "average" cooling demand defined by the duty cycle in Figure 1, the "Average" power to drive the fan is 1.69 HP.	Belt Driven Fan, HP	Fixed Pump w/ Fixed RV, HP	Fixed Pump w/ Prop RV, HP	Tandem Pump w/ Prop RV, HP	Variable Fan Drive Pump, HP
"Average" Engine Power Required by Fan Drive, HP =	8.0	12.6	5.6	3.6	2.7
Waste Power available to heat the hyd. Fluid, HP =		10.9	3.91	1.86	1.04
Net Wasted Power, HP =	6.3	10.9	3.9	1.9	1.0
Estimated Cost of #2 Diesel Fuel, $/Gal =	2.00				
Estimated Brake Specific Fuel Consumption, lb/HP-hr =	0.35				
Average Specific Density of #2 Diesel Fuel, lb/gal =	6.93				
Average Work Cycle, hours/year =	600				
Net Average Fuel Consumed by Wasted Power, Gal/year =	194	334	120	57	32
Average Cost of Wasted Fuel, $/year =	$ 389	$ 668	$ 240	$ 114	$ 64
Cost Comparison: Hydraulic Fan Drive vs Belt Drive					
Net Additional Cost of Fixed Pump w/ Fixed RV, $/year =		$ 279			
Net Cost Savings; Proportional Hydraulic vs Belt Drive, $/year =			$ 149	$ 275	$ 325
Average CO_2 Production per Gallon of Diesel Fuel, lb/gal =	22.40				
Additional CO_2 Production per year, lb/year =	4351	7481	2684	1277	714

Table 1. Comparison of a belt-driven fan with hydraulic fan drives.

System Design T, ºF =	100				
System Coolant Design Temp, ºF =	185	195	205	215	225
Fan Drive System Overall Efficiency, %	Fan Drive system Ambient "Break-Even" Temperature, ºF (For Cooling System Load at Maximum				
90%	81	91	101	111	121
85%	79	89	99	109	119
80%	77	87	97	107	117
75%	75	85	95	105	115
70%	72	82	92	102	112
65%	70	80	90	100	110
60%	66	76	86	96	106
55%	63	73	83	93	103
50%	59	69	79	89	99

Table 2. For any proportional hydraulic fan drive system operating at maximum heat rejection, there is a break-even ambient temperature, below which the proportional hydraulic fan drive will require less power from the engine.

will be reduced. Based on the Fan Law Equations, the required fan power can be reduced further by: $(HP_1/HP_0) = (HR_1/HR_0)^3$.

Engine heat rejection is likely to increase with the implementation of the emissions regulations. Thus, reducing hydraulic losses can offset these increases and may permit continued use of existing radiators, coolers, fans and shrouds in the upgraded machines.

Another advantage is that hydraulic fan drives can be positioned anywhere on the machine. Relocating the cooling system out of the engine compartment can make room available for additional components required for exhaust aftertreatment. Hydraulic fan drives can be located in areas of clean air around the machine. Further, hydraulic fan drives are reversible — dirt and debris can be automatically flushed out of the coolers to ensure that they are operating at peak efficiency.

One of the most often asked questions is, "Isn't the hydraulic fan drive system less efficient than the belt drive?" The answer is a qualified "yes." Yes, the belt drive is more efficient in transmitting mechanical power, but in a fan drive this occurs only when the vehicle is operating at both maximum heat rejection and at or near the maximum design ambient temperature.

For any given proportional hydraulic fan drive system operating at maximum heat rejection, there is a break-even ambient temperature below which the proportional hydraulic fan drive will require less power from the engine. This relationship is illustrated in Table 2.

For a proportional hydraulic fan drive system that has been designed for a nominal 100°F temperature differential between the hottest coolant and the highest ambient temperature, the break-even ambient temperature is related to the combined overall efficiency of the fan drive's hydraulic pump and motor. For the purpose of this illustration, if we assume that the overall efficiency of the hydraulic drive is 70%, then the break-even ambient temperature for a system with maximum coolant temperature of 205°F is 92°F.

Normally, we expect that the cooling system is designed to handle the maximum heat rejection for the entire population of vehicle systems in every climate condition. It is appropriate for the system designer to evaluate whether the two conditions of maximum heat rejection and maximum ambient temperature can occur simultaneously for a significant portion of the machine population. For the example shown in Table 2, the evaluation should include whether a significant portion of the machines will experience maximum heat rejection at ambient temperatures above 92°F.

With new global emissions regulations on the horizon, we should expect that engine coolant temperatures will increase by 5° to 10°F. If the heat rejection and the cooling system remain the same and the coolant temperature increases by 10°F, it can be seen that the break-even ambient temperature also increases by 10°F to 102°F. If the coolant temperature increases by 10°F, but the design temperature differential is increased by the same 10°F, then it can be shown that the break-even ambient temperature would remain at approximately 92°F. dp