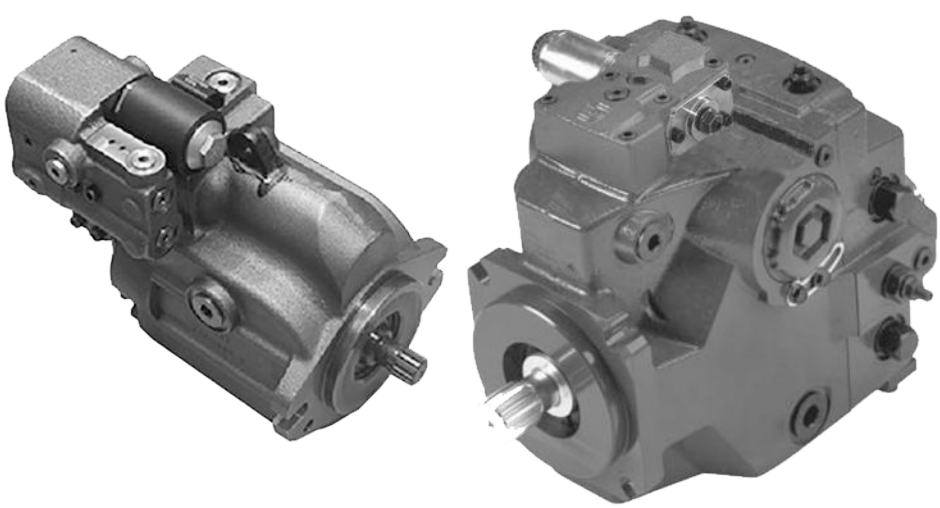
Design Guidelines

Hydraulic Fan Drive Systems



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Revision history

Table of revisions

Date	Changed	Rev
January 2020	Updated Appendix H with new graphs for all frame sizes	0401
July 2018	Appendix H - new graphs for some frame sizes	0303
May 2018	add notes: H1 pumps with fan drive control	0302
April 2017	Updated Appendix H chapter	0301
July 2015	Danfoss layout - Add Appendix I - RDM Fan Drives	0201
2006 - 2013	Various changes.	AA up to BC

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Introduction

Abstract

Fan drive system sizing relies heavily on the input received from the customer. All system sizing calculations are based on the required fan power @ trim speed data given to the hydraulic system design engineer. This data is a statement of the fan drive motor shaft power that is required to turn a fan at the required speed to push, or pull, a required volume of air across coolers/radiators. The usual sequence of events is:

- The engine manufacturer advises the customer, or cooling system designer, of the heat dissipation required from the cooling system, charge air cooler etc. This information is combined with the heat rejection data for any accessories and work functions on the machine (such as : transmission cooler, hydraulic cooler, and A/C condenser) to determine the maximum heat rejection profile for the system.
- The customer's cooling pack manufacturer uses this data to size the cooling package and generally recommends a fan to suit this need, providing the rated fan power, rated fan speed, and the fan speed and static pressure required to satisfy the cooling needs of the system.
- With this information, knowing the minimum engine speed at which maximum fan speed needs to occur, the hydraulic system designer can size the hydraulic fan drive system.

Overview

One goal of this document is to provide the reader with the equations and formulae needed to size a hydraulic fan drive, given that they are provided with the following information:

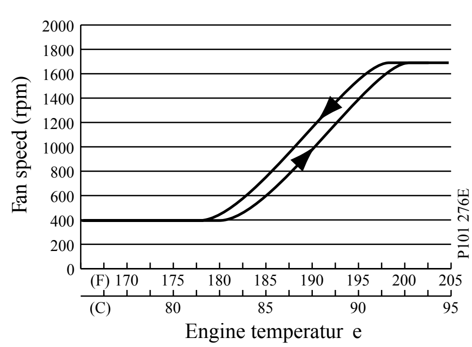
- Rated fan power.
- Rated fan speed.
- Fan speed required to meet the maximum cooling needs of the cooling system.
- Engine speed at which maximum system cooling is required.

This document also provides an explanation of the terms and factors used in the derivation of the sizing equations. In addition, the reader is provided with recommendations of simple system design solutions that will help provide a viable system with satisfactory performance.

Principles of Operation

The vehicle's cooling fan is driven by a hydraulic motor, which in turn, is driven by a hydraulic pump. The hydraulic pump can be driven directly off of the engine supplied PTO (Power Take Off), or with a belt drive. An electrically controlled proportional pressure control valve modulates the fan speed depending on a temperature reading. In a cold condition, the fan idles with very low power consumption. During the hot condition, the maximum fan speed is controlled by a pressure control valve, which adjusts the fan speed to meet the cooling needs of the total system. Every system has a temperature, which allows for the most efficient performance. The electronic control system, attempts to maintain the coolant at the optimum design temperature, which the "system integrator" selects during the design phase of the project.

Fan speed vs engine temperature



To optimize the cooling system operation in various environmental conditions and to minimize parasitic losses, the Danfoss modulating fan drive system enables the fan cycle to be designed to specific heat

Introduction

rejection requirements for a wide range of environmental conditions. Vehicle manufacturers have complete control of the fan cycle by choosing the appropriate temperature limits.

The Danfoss modulating fan drive system remains at idle speed until conditions require increased fan speeds. By regulating the pressure drop across the hydraulic motor, modulation of fan speed occurs, and over-cooling is prevented.

Power Savings

In the **fan off** condition, the fan may idle at approximately 30% of rated speed, but it will only consume about 3% of rated power. The Danfoss modulating fan drive system allows the system designer to size the fan for the engine speed at which maximum heat rejection occurs. The fan speed will remain essentially constant at all higher engine speeds. Consequently, the fan will not require excessive parasitic losses as engine speed increases. In systems where the engine speed at maximum heat rejection is 80% of the governed speed, the power savings compared to over-speeding a mechanically driven fan can be as high as 95%.

Modulation Preferred Over on/off Fan Speed Control

Fan speed modulation occurs over a temperature range chosen by the system's designer. This eliminates the sudden changes in speed that cause dramatic changes in noise levels. Similarly, large accelerations of components, which may limit the reliability for long-term operation, are eliminated. Modulation also allows intermediate levels of cooling without unnecessary cycling of the fan between minimum and maximum speed. The calibration temperature, operating range, and ramp times can be varied independently by the system designer to achieve the desired level of temperature control.

Fan Drive Design

Design Considerations

- Parasitic losses from excessive fan speed are high. Power consumed by a fan is proportional to fan speed cubed ($speed^3$).
- Heat rejection to the atmosphere does not increase linearly with engine speed.
- Overheating and/or over cooling the system will result in loss of efficiency and productivity.
- Overheating, and/or over cooling the engine can result in increased emissions to the environment.
- The proportion of operating time during which full fan speed (maximum fan power) is needed is typically about 20% and can be as low as 5%.
- Mounting the fan directly to the engine requires large fan blade tip clearances due to normal engine vibration and movement. This leads to loss of fan performance. Mounting the fan directly to a hydraulic motor can minimize tip clearance and boost fan performance significantly.

Hydraulic fan drive system designers select components for unique combinations of engine, fan, and application parameters. Do not exchange/change fan system components indiscriminately. Design factors which determine the selection of the fan drive system for a particular engine, or vehicle, include:

- Engine set point and maximum heat rejection
- Pump rotation
- Pump input torque limitations
- Maximum applied pressure and speed limits for the individual components
- Fit-up and available installation space
- Pump support structure requirements for individual engine mounting combinations
- Specific engine and accessory temperature control limits

Fan drive element selection

Fan drive Element	Design parameter	Design flexibility	Design champion
Engine & accessories	Power, speed, total heat rejection and duty cycle	Yes	OEM
PTO & Gearing	Engine to pump gear ratio	Sometimes	OEM's choice of engine supplier
Pump(s)	Displacement, pressure, speed, fixed pump or variable pump, mounting & drive line	Yes	Danfoss technical representative, & OEM
Fan drive control	Sensor input(s), control output, number of control elements	Yes	Danfoss technical representative, & OEM
Motor(s)	Displacement, pressure, speed, fixed motor or variable motor, mounting & coupling	Yes	Danfoss technical representative, & OEM
Fan(s)	Fan rated power @ rated speed, fan diameter, number of blades, blade pitch, proximity of blades to heat exchanger, direction of air flow	Yes	OEM & cooling specialists
Shroud(s)	Type of shroud (flat plate, short duct, venturi), blade axial position in shroud, tip clearance	Yes	OEM & cooling specialists
Air flow stream(s)	Air flow rate & static pressure across heat exchanger(s), maximum ambient air temperature, minimum atmospheric pressure, hot air recirculation, baffles, louvres & obstructions	Sometimes	OEM & cooling specialists
Heat exchangers	Physical size, height & width, number of heat exchangers in air flow stream, side-by-side, axial stack, materials selected, construction, number and types of tubes, tube configuration, fin density	Yes	OEM & cooling specialists

Fan Drive Components

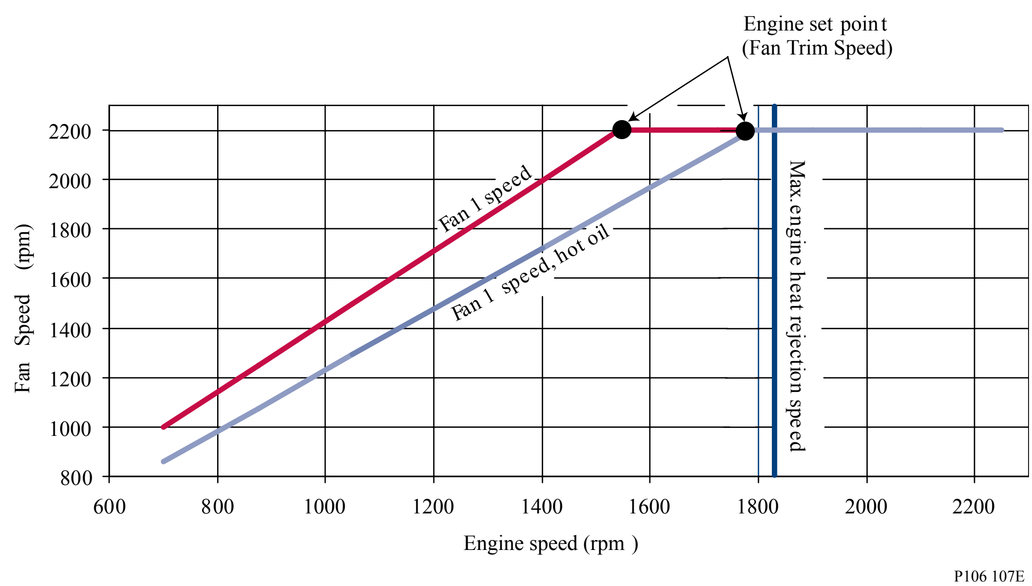
Fan Drive Element Selection

Optimizing the size of fan drive elements depends on selecting the correct components and gear ratios. By matching these components to the fan power requirements, the required unit sizes can be quickly determined. The pump and motor displacements, input gear ratios, engine set point, and pressure limits can be adjusted to provide some optimization of component size. Along with the sizing equations presented in this article, a Danfoss fan drive sizing computer tool is available to assist with sizing the hydraulic components.

Many modulating hydraulic fan drives rely on dedicated pumps to provide flow to the fan circuit for optimum sizing. Other circuits are available that provide additional flow for power assisted steering and other accessory systems. In these, and many other circuits, the sizing equations and fan drive sizing tool may still be used to select the required components. Note that the design limits for associated design elements are not identified in this article. They may be reviewed by referring to the Danfoss technical information for the components being considered. Machine designers should verify that all design parameters are met for all drive line components.

While the methods described in this article may be useful, they do not represent the only approach to sizing hydraulic components. Contact your Danfoss representative if questions of interpretation exist.

Collect the application sizing parameters as identified in the System Design Parameters chapter of this document. Pay particular attention to the minimum engine speed at which maximum heat rejection to the atmosphere is required. When sizing the pump for the application, the system designer should ensure that the engine set point under hot oil condition is less than the engine speed at which maximum heat rejection occurs. Failure to do this can result in a condition where the cooling system may not provide adequate cooling when maximum work loading and maximum ambient conditions occur simultaneously.



Sample graph, performance prediction will vary depending on choice of input parameters.

One of the first things that the systems designer should consider is whether the maximum pump torque needed will exceed the input torque limitation of the pump drive. One way to calculate this is to divide the fan power requirement by an estimate of the hydraulic system's overall efficiency and then determine the input torque requirement at the pump speed that is equivalent to the maximum heat rejection speed of the engine.

Estimate of Maximum Input Torque to the Pump

Compare the estimated maximum input torque to the maximum available input torque at the pump drive; this will determine the design margin that is available to the designer. The hydraulic system designer should consult with the vehicle system's designer, and/or the prime mover's technical support staff for assistance, if required.

Fan Drive Components

SI system

$$\text{Pump torque, } Tp_{(N\cdot m)} \approx \left[\frac{\left(\frac{Pf_{kW}}{0.7}\right)\cdot(9549.0)}{(Ne \cdot R)}\right] \quad (N\cdot m)$$

English system

$$\text{Pump torque, } Tp_{(lbf\cdot in)} \approx \left[\frac{\left(\frac{Pf_{hp}}{0.7}\right)\cdot(63025)}{(Ne \cdot R)}\right] \quad (lbf \cdot in)$$

Where:

Pf_{kW}	= Max fan power, kW [hp]
(Ne • R)	= Pump speed, rpm
R	= Pump/Engine ratio
Ne	= Engine speed, rpm
0.7	= η_t for hydraulic system (pump and motor)

Converting terms

$$\text{Pump torque, } Tp_{(lbf\cdot in)} = Tp_{(N\cdot m)} \cdot 8.8507 \quad (lbf\cdot in)$$

$$\text{Pump torque, } Tp_{(lbf\cdot ft)} = Tp_{(N\cdot m)} \cdot 0.7376 \quad (lbf\cdot ft)$$

$$\text{Pump torque, } Tp_{(N\cdot m)} = \left(\frac{Tp_{(lbf\cdot in)}}{8.8507}\right) \quad (N\cdot m)$$

$$\text{Pump torque, } Tp_{(lbf\cdot ft)} = \left(\frac{Tp_{(lbf\cdot in)}}{12.0}\right) \quad (lbf\cdot ft)$$

System Design Parameters

Sizing

Fan drive system sizing relies heavily on the input received from the customer. All system sizing calculations are based on the required fan power @ trim speed data given to the hydraulic system design engineer. This data is a statement of the fan drive motor shaft power that is required to turn a fan at the required speed to push, or pull, a required volume of air across coolers/radiators. The usual sequence of events is:

- The engine manufacturer advises the customer, or cooling system designer, of the heat dissipation required from the cooling system, charge air cooler etc. This information is combined with the heat rejection data for any accessories and work functions on the machine (such as : transmission cooler, hydraulic cooler, and A/C condenser) to determine the maximum heat rejection profile for the system.
- The customer's cooling pack manufacturer will then use this data to size the cooling package and will generally recommend a fan to suit this need, providing the rated fan power, rated fan speed, and the fan speed required to satisfy the cooling needs of the system.
- With this information, knowing the minimum engine speed at which maximum fan speed needs to occur, the hydraulic system designer can size the hydraulic fan drive system.

To completely understand any fan drive system is to understand the fan load characteristics. Fans are unique in that the power to drive the fan changes with the cube of the fan speed, as follows:

$Pf = k \cdot (N_f)^3$

$Pf_1 / Pf_2 = (N_{f1} / N_{f2})^3$

Where:

Pf = fan power (kW, hp)

N_f = fan speed (rpm)

1,2 = subscripts for two different conditions

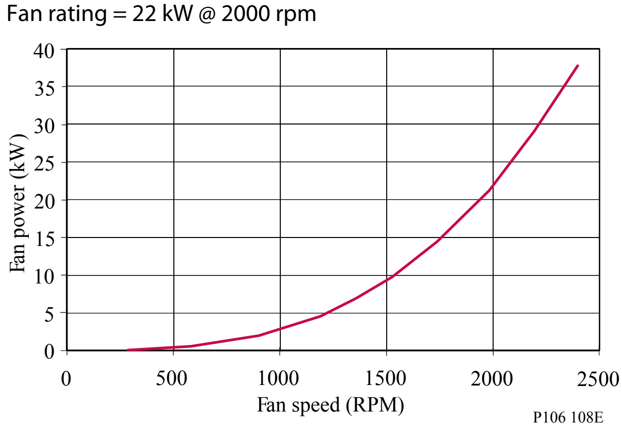
k = Fan power coefficient

Fan power is defined as the power required to drive the shaft connected to the fan and is equal to the output power of the motor.

When a given fan speed is doubled; the required power to drive the fan increases by a factor of 8.

Fan power requirements (Example)

Fan rating = 22 kW @ 2000 rpm



Since fan power is a function of both pressure and flow (fan speed), it follows that the relationship between fan speed and system pressure is

$\Delta P_1 / \Delta P_2 = (N_{f1} / N_{f2})^2$

Where: ΔP = delta pressure across the hydraulic motor (bar, psid)

An accurate value of the fan rating is critical to the correct selection of components and their settings.

System Design Parameters

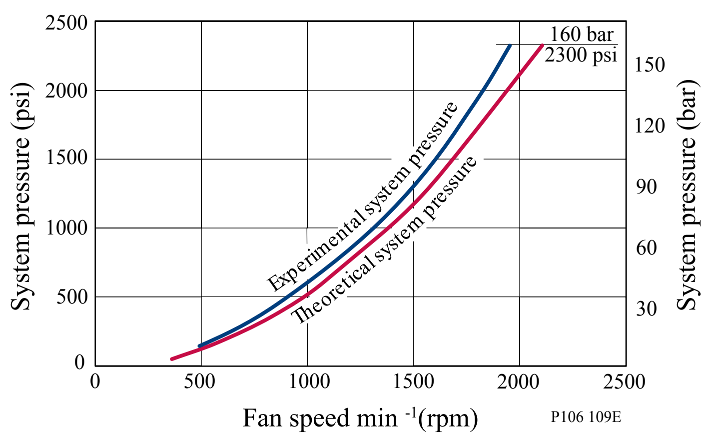
Although the cubic relationship between fan power and fan speed has been consistently verified experimentally, it is still an approximation of the fan behavior. Therefore, to avoid significant errors in predicting power requirements, the fan power rating should be taken at a speed representative of typical fan operation.

For example, for a system in which the fan usually operates in a speed range of 1800-2200 rpm, a fan rating specified at 2000 rpm will yield more accurate results than a rating specified at, say, 1500 or 2500 rpm.

Fan curves provided by the fan manufacturer are often developed under ideal conditions. It is unlikely that a fan will exhibit exactly the same performance in an actual application (because of: shrouding, heat exchange airflow characteristics, and air density). Only through test data taken on the actual vehicle can a fan's performance characteristics be accurately determined. The curve below illustrates the differences between predicted performance and actual performance of a fan installed in a vehicle. The system designer/integrator is encouraged to confirm their performance predictions via test over the entire operating speed range of the engine, and to refine their prediction model with a revised *fan power coefficient* when they rerun the sizing calculations.

Fan power requirements (example)

Fan rating = 22 kW @ 2000 rpm



System Design Parameters

Sizing Equations

Equations

Pumps

Based on SI units

Output flow $Q = \dfrac{V_g \cdot n \cdot \eta_v}{1000}$ (l/min)

Input torque $M = \dfrac{V_g \cdot \Delta p}{20 \cdot \pi \cdot \eta_m}$ (N•m)

Input power $P = \dfrac{V_g \cdot n \cdot \Delta p}{600\,000 \cdot \eta_m}$ (kW)

Based on English units

Output flow $Q = \dfrac{V_g \cdot n \cdot \eta_v}{231}$ (US gal/min)

Input torque $M = \dfrac{V_g \cdot \Delta p}{2 \cdot \pi \cdot \eta_m}$ (lbf•in)

Input power $P = \dfrac{V_g \cdot n \cdot \Delta p}{396\,000 \cdot \eta_m}$ (hp)

Motors

Based on SI units

Output torque $M = \dfrac{V_g \cdot \Delta p \cdot \eta_m}{20 \cdot \pi}$ (N•m)

Output power $P = \dfrac{Q \cdot \Delta p \cdot \eta_t}{600}$ (kW)

Based on English units

Output torque $M = \dfrac{V_g \cdot \Delta p \cdot \eta_m}{2 \cdot \pi}$ (lbf•in)

Output power $P = \dfrac{Q \cdot \Delta p \cdot \eta_t}{1714}$ (hp)

Variables

SI units [English units]

V_g = Displacement per revolution cm^3/rev [in^3/rev]

p_O = Outlet pressure bar [psi]

p_i = Inlet pressure bar [psi]

Δp = pO - pi (system pressure) bar [psi]

n = Speed min-1 (rpm)

η_v = Volumetric efficiency

η_m = Mechanical efficiency

η_t = Overall efficiency ($\eta_v \cdot \eta_m$)

SI unit formulas are based on cm^3, bar, N, N•m, W.
English formulas are based on in^3, psi, lbf•in, hp.

System Design Parameters

Axial Flow Fan Power Formula

Power to system parameter relationships

$$\frac{Pf_2}{Pf_1} = \frac{(N_2)^3 \cdot (D_2)^5 \cdot \nu_2}{(N_1)^3 \cdot (D_1)^5 \cdot \nu_1}$$

$$\frac{Pf_2}{Pf_1} = \frac{r_2}{r_1} \cdot \left(\frac{N_2}{N_1}\right)^3 \cdot \left(\frac{D_2}{D_1}\right)^5 = \frac{\nu_2}{\nu_1} \cdot \left(\frac{N_2}{N_1}\right)^3 \cdot \left(\frac{D_2}{D_1}\right)^5$$

$$\frac{\dot{V}_2}{\dot{V}_1} = \left(\frac{D_2}{D_1}\right)^3 \cdot \left(\frac{N_2}{N_1}\right)$$

$$\frac{\Delta P_2}{\Delta P_1} = \frac{r_2}{r_1} \cdot \left(\frac{D_2}{D_1}\right)^2 \cdot \left(\frac{N_2}{N_1}\right)^2 = \frac{\nu_2}{\nu_1} \cdot \left(\frac{D_2}{D_1}\right)^2 \cdot \left(\frac{N_2}{N_1}\right)^2$$

Pf_1 = Power of fan at known condition #1

Pf_2 = Power of fan at condition #2

N_1 = Fan speed at condition #1

N_2 = Fan speed at condition #2

D_1 = Fan diameter at condition #1

D_2 = Fan diameter at condition #2

ν_1 = Specific weight of air at condition #1

ν_2 = Specific weight of air at condition #2

r_1 = Density of air at condition #1

r_2 = Density of air at condition #2

ΔP_1 = Hydraulic and/or Static Pressure at condition #1

ΔP_2 = Hydraulic and/or Static Pressure at condition #2

V_1 = Flow rate of air at condition #1

V_2 = Flow rate of air at condition #2

System Design Parameters

System Design Data Form

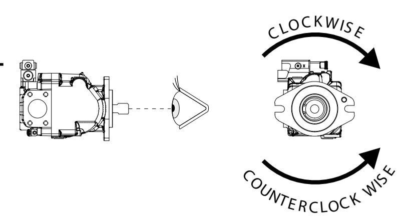
Print this form. Fill in all the fields and check the appropriate check boxes. Fax the filled out form to your Danfoss Power Solutions Technical Sales Representative.

Engine details

Manufacturer ______________ Model or Series __________________

Pump Drive
Engine PTO ☐ Ratio ______ :1 (engine to pump) Input torque limit: ______
Belt Drive ☐

Pump Rotation
Clockwise, Right hand ☐
Counterclockwise, Anti-clockwise, Left hand ☐



Speeds
Low Idle __________ RPM (rated)
Governed __________ RPM (rated)
High Idle __________ RPM (max speed)

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Power steering

(if applicable)

Controlled Flow Requirement ____________ US gal/min ☐ l/min ☐

Steering Pressure (maximum) ____________________ psi ☐ bar ☐

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System Design Parameters

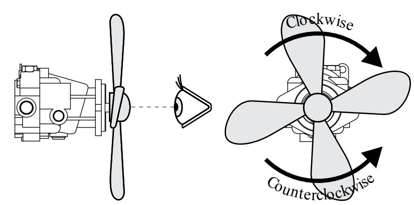
Fan information

Manufacturer ____________ Model or Series ____________

Fan Diameter ____________ in ☐ mm ☐

Fan Input Power ____________ HP ☐ kW ☐ At speed ______ rpm

Fan Rotation **Clockwise** ☐
(viewed on motor shaft, see illustration) **Counterclockwise** ☐



Fan Trim Speed ____________ rpm

Set Point at Fan Trim Speed ____________ rpm
(engine speed where max heat load occurs)

Coolant Temperature at Fan Trim Speed ______ **°F** ______ **°C**
(coolant temp where max fan speed is required)

Note: To properly size and specify a fan drive system, fan power requirements must be stated as accurately as possible. Fan power requirements can be determined from fan curves supplied by the manufacturer. Radiator and cooler manufacturers will supply air flow requirements based on heat loads. Air flow information must include accurate air flow and static pressure to determine correct fan power requirements.

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Control preference

	Electro-Hydraulic Modulating	**Electro-Hydraulic ON/OFF**
Single Input	☐	☐
Multiple Inputs	☐	☐

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Reservoir

Reservoir Capacity ____________ US gal ☐ liter ☐

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System Design Parameters

Fluid

Hydraulic Fluid Type ____________________

Viscosity ____________________ at 40° C [104°F] ☐ cSt ☐ SUS

____________________ at 100° C [212°F]

Maximum Fluid Temperature ____________________ °C ________ °F

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Filtration

Filter Position
- Inlet Line ☐
- Pressure Line ☐
- Return Line (recommended) ☐

Filter Flow
- Full Flow ☐
- Partial Flow ☐

Filter Rating ____________________ micron ____________ x ratio

Note: Do not locate the filter cartridge inside the reservoir. This reduces the reservoir capacity and reduces the dwell time (the time the oil spends in the resrevoir). It also increases the potential for damage to the hydraulic components due to aeration of the oil.

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System Design Parameters

Technical Features

In this document, we introduce the equations that are used to size the components of a modulating fan drive system. In addition to these principles, there are several other factors to consider to ensure that the hydraulic system performs to expectations. Following are some considerations you are encouraged to address during the design phase.

Shaft Loads and Bearing Life

For information on shaft loads and bearing life, refer to:

- Series 45 Technical Information **520L0519**
- Series 40 Motors Technical Information **520L0636**
- Series 42 Pumps Technical Information **11022637**
- H1 Pumps Technical Information Manuals (see *Reference Literature* at the back of this manual)

Maximum Pump Speed

Pump displacement, and centrifugal filling of the pumping chambers, limit maximum pump speed. Unless otherwise specified, maximum rated pump speeds are based on operation at sea level with hydraulic fluids having a specific gravity of 0.9 and a viscosity of 58 SUS (9 cSt) at 180° F (80° C). Speed limits for a particular application depend on the absolute pressure and oil viscosity. Speed limits for individual products may be found in their respective technical information bulletins. Consult a Danfoss representative for operation outside of these published limits.

Minimum Pump and Motor Speed

Volumetric efficiency limits minimum pump speed. If lower than recommended starting or operating speeds are required, contact a Danfoss representative for assistance. Piston motors are designed for continuous operation at low speed, and at rated pressure. Motors may be started from zero speed on fan drives, and torque will increase with speed.

Motor Starting Pressure (open circuit motors)

No-load motor start-up pressures may range from 100 to 725 psid (7 to 50 dbar), depending on displacement. This property of the motor is dependent on motor design parameters, the CSF (Coefficient of Static Friction), and it is inversely proportional to motor displacement. For example: The starting torque for any given motor is largely dependent on the pitch diameter of the pistons and the CSF. Since torque is dependent on the product of pressure and displacement, and starting torque is essentially constant for any given frame size; starting pressure will be dependent on displacement, in an inverse relationship. To minimize starting pressure, select the smallest frame size for the required motor displacement.

Besides displacement, there are several factors which also effect motor starting pressure. They include: pressure rise rate (pressure gradient), temperature, fluid viscosity, motor return pressure (back pressure), fan inertia, pump flow rate, and piece-to-piece variation between motors.

Motor Free Run Pressure

Free run pressure is the minimum delta pressure across the motor that is required to keep the motor turning when there is no cooling demand. Free run pressure is dependent on motor displacement and shaft speed.

If the delta pressure across the motor is allowed to fall below the free run pressure; the motor will stop, and it will be necessary for the motor to go through the starting condition (start-up procedure) when cooling is needed again. In most applications, it is desirable to initiate fan rotation when the engine starts and prevent the motor from stopping, while the engine is running.

System Design Parameters

Input Torque Ratings

When applying pumps in multiple configurations, ensure the input torque limitations are met for each section and for cumulative sections. Refer to individual product technical information bulletins for specific product torque limits. Always ensure that any individual pump in a multiple unit does not exceed its respective torque rating.

Caution

Torques in excess of recommended values may cause premature input shaft, or unit, failure.

Pump Drive Conditions

Most Danfoss products are available with SAE and metric, standard spline, tapered key, or cylindrical keyed drive shafts for direct or indirect drive applications. An intermediate coupling is the preferred method for direct drives, thereby eliminating radial and axial loading. Direct Drive (or *plug-in* or *rigid*) spline drives can impose severe radial loads on the pump shaft when the mating spline is rigidly supported. Increased spline clearance does not alleviate this condition. Both concentricity and angular alignment of shafts are important to pump life. Misalignment can induce excessive side loads on bearings and seals, causing premature failure.

Overhung load drives (chain, belt, or gear) are permissible. Contact Danfoss for assistance. The allowable radial shaft loads are a function of the load magnitude, the load position, the load orientation, and the operating pressure of the hydraulic pump. All external shaft loads will have an effect on bearing life and may affect pump performance. In applications where external shaft loads cannot be avoided; optimizing the position, orientation, and magnitude of the radial load can minimize their influence on the pump. A tapered input shaft is recommended for applications where radial shaft loads are present. (Spline shafts are not recommended for belt or gear drive applications, the clearance between the mating splines will prevent accurate alignment of the drive elements and will contribute to excessive wear of the spline.) For belt drive applications, a spring loaded belt-tensioning device is recommended to avoid excessive radial loads on the input shaft.

Note for H1 pump with an FDC: Due to the failsafe functionality of the H1P FDC control the pump will stroke to max. displacement in case the input signal to the pump control and the Diesel engine will be switched off at the same time. In this situation a low loop event can occur which may damage the pump. Therefore, it’s strictly recommended to keep the input signal to the pump control alive while switching off the engine.
For further information please contact your Danfoss representative

Tapered Shaft and Hub Connections

Tapered shaft/hub connections provide excellent control of both axial and radial position of the drive coupling or fan assembly. When using the tapered connection, additional effort should be used to insure that there is adequate axial clamping load between the hub and the shaft. The designer is encouraged to establish that there is:

- Adequate clearance under the bolt/nut to insure full axial load may be applied to the taper without bottoming out.
- Adequate clearance between the top of the key and the bottom of the keyway in the hub. Interference between the top of the key and the bottom of the keyway will prevent the hub from seating onto the taper of the shaft. This will compromise the ability of the shaft to transmit its full torque capacity, and may result in failure of the shaft.

Pump Suction

For maximum pump life, the inlet pressure should not drop below 0.8 bar absolute [6 in. Hg vac.] at the pump inlet port.

For cold start conditions, inlet pressure down to 0.6 bar absolute [12 in. Hg vac.] is acceptable for short durations. The possibility of damage due to fluid cavitation and aeration is proportional to decreases in

System Design Parameters

inlet pressure. In addition, oil film lubrication may be disrupted by low inlet pressure. These factors, either singularly or combined, may contribute to a decrease in pump life. Multiple changes in either diameter or direction can have a significant effect on the resistance to flow in inlet passages and can result in a substantial increase in the *effective* length of the inlet line. For this reason, Danfoss recommends that the inlet line contain a minimum number of adaptor fittings, tees, and elbows; as each are a source of additional restriction and, potentially, a source of leakage.

Caution

Continuous operation with inlet pressures below 0.8 bar absolute [6 in. Hg vac.] can cause premature unit failure. Ensure adequate flow/pressure head at the pump inlet at all times.

Case Drain Pressure

Maximum pressure limitations for both case drain and inlet passages are available by consulting the appropriate technical information bulletin for the products being applied. Both line length and diameter influence the pressure drop of the fluid in these passages as it flows to/from the reservoir. In addition, both steady state flow velocity and transient conditions, which can accelerate the fluid in these passages, must be considered when determining their correct size. Of the two design parameters: line length and diameter, diameter has the most influence on the success of the design. Increasing line diameter can decrease both the steady state and the transient pressure drops exponentially. For additional information on steady state pressure drops in hydraulic passages, the reader is encouraged to consult any good text on basic hydraulic design. For additional information on transient pressure drops, refer to Appendix D.

Introducing additional flow from external sources into these return lines can also result in transient pressure pulses that may exceed the drain, or case pressure limits of these products. Danfoss recommends that the bearing drain and case drain lines return directly to the reservoir and remain dedicated to their intended function without connecting them to additional flow sources.

Filtration

To prevent premature wear, it is imperative that only clean fluid enters the pump and hydraulic circuit. A filter capable of controlling the fluid cleanliness to class 22/18/13 (per ISO 4406-1999) or better, under normal operating conditions, is recommended. At initial start up, the system can be at Class 25/22/17 but should not be run at high speed or pressure until the Class 22/18/13 is achieved through filtration. Since the filter must be changed at regular intervals, the filter housing should be located in an accessible area. Appropriate filter change intervals may be determined by test or by gauges indicating excessive pressure drop across the filter element.

For more information refer to *Design Guideline for Hydraulic Fluid Cleanliness, Technical Information* **520L0467**.

Operating Temperatures

With Buna seals and normal operating conditions, the system temperature should not exceed 82 °C [180 °F] except for short periods to 93 °C [200 °F]. With optional Viton elastomer, the system may be operated at continuous temperatures up to 107°C [225°F] without damage to the hydraulic components.

Caution

Operation in excess of 107 °C [225 °F] may cause external leakage or premature unit failure.

Fluids

A mineral based fluid is recommended that includes additives to resist corrosion, oxidation and foaming. The oil should have a maximum viscosity commensurate with system pressure drop and pump suction pressures. Since the fluid serves as a system lubricant, as well as transmitting power, careful selection of the fluid is important for proper operation and satisfactory life of the hydraulic components. Hydraulic

System Design Parameters

fluids should be changed at appropriate intervals determined by test, supplier, or by change in color, or odor, of the fluid.

Every 10°C [18°F] rise in continuous reservoir temperature over 80°C [176 °F] decreases the life of the oil by ½.

For additional technical information on hydraulic fluids refer to *Hydraulic Fluids and Lubricants* **520L0463** Technical Information Bulletin and specific product technical bulletins.

For information relating to biodegradable fluids, see Danfoss publication *Experience with Biodegradable Hydraulic Fluids* **520L0465** or consult the Danfoss Technical Services Department.

Mounting

The pump mount/drive should be designed to minimize axial and radial loads on the shaft. When using an indirect (chain, belt, or gear) drive, contact Danfoss to determine permissible load limits and orientation of the installation.

The motor mount should be designed to position the motor/fan assembly within the shroud for optimum fan performance and to locate the leading edge of the fan blades relative to the adjacent surface of the heat exchanger. The support structure should be constructed so that it will be robust against forces and deflections due to shock and vibration as well as the loads applied to it by the fan and the hydraulic plumbing that will be connected to the motor.

Axial Thrust Motors

When a fan is directly mounted onto the drive shaft of a hydraulic motor, it imparts both a radial and an axial thrust load onto the shaft. In general, the weight of the fan is insignificant when compared to the radial load capacity of the bearings in the motor. But, the axial thrust load must be considered carefully. Under normal operating conditions, Danfoss motors have adequate axial thrust capacity for most fans that are applied in the industry, but they do have limitations. It is recommended that the system designer determine the axial thrust force that will be produced by the fan and compare it to the values listed below:

Series 40 motors external shaft load limits

	Unit	**M25**	**M35/44**	**M46**
M_e	N•m [lbf•in]	29 [256]	25 [221]	24 [212]
T	N [lbf]	848 [190]	966 [217]	1078 [242]

L and K motors external shaft load limits

Frame		L		K	
Mounting configuration		SAE	Cartridge	SAE	Cartridge
Maximum allowable external moment (Me)	N•m	7.7	21.7	13.3	37.5
	[lbf•in]	68	192	118	332
Maximum allowable thrust load (T)	N	750		1100	
	lbf	169		247	

Refer to Appendix-A for equations that will assist in calculating the axial thrust loads from the fan. Calculated loads should be confirmed by test.

For shaft load limit calculations on Series 90 motors and H1B motors, contact your Danfoss representative.

System Design Parameters

Piping

The choice of piping size and installation should always be consistent with minimizing maximum fluid velocity. This will reduce system noise, pressure drops and overheating, thereby adding to cost savings for the operation of the system. Inlet piping should be designed to prevent continuous pump inlet pressures below 0.8 bar abs. [6 in. Hg vac.] or 0.6 bar abs. [12 in. Hg vac.] during start-up. When selecting pipe sizing, recognize pressure drops are related to flow velocity. Danfoss recommends limiting the maximum average mean flow velocity to 5 m/sec [15 ft/sec.] in pressure lines, and 2.5 m/sec [7 ft/sec.] in suction lines.

In addition to limiting maximum flow velocity, it is recommended that the designer select the hoses, fittings and integral valve elements to be compatible with the desired working pressure of the hydraulic system. The following documents may be used to determine the working pressure ratings for the respective system elements:

- SAE J514: for working pressure ratings and fitting installation torques for - O-ring boss fittings/ports and JIC 37° flared tubing connections
- SAE J518: for working pressure ratings and bolt installation torques for SAE code 61 4-Bolt flange fittings/ports,
- SAE J517: for working pressure ratings for SAE hydraulic hose
- SAE J1453: for working pressure ratings for flat face O-ring fittings.

Reservoir

The reservoir should be designed to accommodate expected maximum volume exchange during all system operating modes and to prevent aeration of the fluid as it passes through the reservoir. Return and inlet lines should be positioned below the reservoir low oil level and be located as far as possible from each other. A diffuser and a baffle plate located between the pump inlet and return line is desirable to reduce turbulence and to allow the oil to de-aerate before it re-enters the pump.

Reservoirs must be sized to ensure de-aeration of the oil before it re-enters the pump. For dwell times of less than 90 seconds, the system designer is encouraged to verify that entrained air (bubbles) are not included in the oil that is being transmitted from the reservoir to the pump. This may be accomplished by placing a sight gage into the inlet line between the reservoir and the pump. Placing a variable frequency strobe light source behind the sight gage will improve the observer's ability to see air bubbles present in the fluid as it passes through the inlet line.

Danfoss encourages system designers to locate the reservoir so that the oil level in the reservoir will remain above the level of the inlet port of the pump under all conditions. By doing this, a positive head is produced that can offset the effects of line losses and altitude on the inlet pressure available at the pump.

Danfoss also encourages system designers to consider the potential for air to be introduced into the inlet line within the reservoir via the introduction of a vortex or whirlpool, between the surface of the oil and the inlet port. One way to discourage a vortex is to locate a baffle between the inlet passage, or suction strainer, and the surface of the oil. The system designer should consider the design parameters of size and position for the baffle to ensure that a vortex cannot form if the reservoir attitude is at its extremes, the oil level is at or below the minimum recommended capacity, or if sloshing occurs due to operation of the machine.

Cavitation and Aeration Damage

Hydraulic oil used in the majority of systems contains about 10% dissolved air by volume. This air, under certain conditions of vacuum within the system, is released from the oil causing air bubbles. These entrained air bubbles collapse if subjected to pressure, and this collapse creates erosion of the adjacent metal surfaces and degradation of the oil. Because of this, it becomes obvious that the greater the air content within the oil, or the greater the vacuum in the inlet line, the more severe will be the resultant damage. The main causes of over-aeration of the oil are air leaks, particularly on the inlet side of the pump, and flow line restrictions such as inadequate pipe sizes, elbow fittings and sudden changes in flow passage cross-sectional area. To avoid cavitation problems when using Danfoss pumps and motors, avoid defects in plumbing and construction, maintain pump inlet pressure and rated speed requirements, and ensure reservoir size and follow recommended guidelines.

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When entrained air entering the pump is pressurized at the pump outlet, it is forced into solution in the oil as the bubbles collapse. This super-saturated solution of dissolved air and oil will release its air when the pressure is released. Symptoms of this condition can be observed by oil / foam escaping from the fill port of the reservoir when the system is shut down.

Cooling

Depending on duty cycle and reservoir/line construction, an oil-cooler may be required. The oil-cooler size is based on typical power losses in the hydraulic circuit. The oil cooler is usually placed in the return line to the reservoir.

Pressure Protection and Ratings

The pump, as well as other system components, has pressure limitations. Thus a relief valve, or pressure limiting device, must be installed in the system, and its setting must be consistent with the product ratings. Refer to the relevant Danfoss technical bulletins for this information.

Caution

Failure to install a relief valve or over-pressure protection may result in premature unit failure.

Bearing Life Expectancy

All Danfoss piston pumps and motors utilize anti-friction, rolling element bearings, and journal bearings, which have an oil film maintained at all times between the bearing surfaces. If this oil film is sufficiently sustained through proper system maintenance and the product's operating limits are adhered to, a long bearing life can be expected.

A B_{10} type life expectancy number is generally associated with rolling element bearings. Bearing life is a function of speed, system pressure, and other system parameters such as oil viscosity and oil cleanliness.

Glossary

Terminology

Trim speed is the maximum fan speed required at the full-on condition. This is equal to, or greater than, the fan speed required to meet the maximum cooling needs of the cooling system.

Engine set point is the engine speed at which trim speed should occur, and is provided by the cooling system designer. This is equal to, or less than, the engine speed at which maximum system cooling is required.

Fan power at trim is the power that needs to be generated at the motor shaft to drive the fan at trim speed.

Fan rating is the value by which different types of fans can be compared. Usually designated as X power @ Y rpm and equates back to an air volume (mass flow rate) that can be moved per minute at the Y rpm.

To assist with the sizing exercise, Danfoss has developed a sizing tool to perform the necessary calculations. Within the sizing tool, worksheets are provided for both fixed displacement pump/fixed displacement motor, and variable displacement pump/fixed displacement motor hydraulic systems. The sizing tool has been provided to your Danfoss representative.

Refer to the data sheets on pages in the System Design Parameters chapter. When the data on these sheets is complete, calculations can be made to determine the most suitable pump/motor/controller combination for the application based on:

- Pump drive available (torque, shaft, mounting flange, overall space envelope)
- System pressure required
- Additional flow/pressure required from the pump, (for example: steering flow)
- Control type requested by the customer
- Limiting operating parameters of the fan drive family products
- Fit (space envelope)

Contact your Danfoss representative for a report of the performance prediction generated by the fan drive sizing tool.

For systems using axial piston pumps, refer to AE Note 2010-02 for sizing calculations. Contact your Danfoss representative for access to AE Note 2010-02.

Appendix A-Fan Performance

Fans

Fans are generally divided into two classifications:

- Centrifugal or radial flow - in which the air flows radially thru the impeller within a scroll type of housing
- Axial flow - in which the air flows axially thru the impeller within a cylinder or ring.

The typical axial flow fan is commonly referred to as a propeller fan, and is customarily used for free delivery, or against low resistance. They are usually mounted within a circular ring or shroud with a circular opening.

Fan Performance

Fan performance is a measure of volume, total pressure, static pressure, speed, power input, mechanical efficiency, and static efficiency, at a stated density. Some useful definitions are:

Volume delivered by a fan is the number of cubic feet of air per minute (or, cubic meters per second), expressed at fan inlet conditions.

Total pressure is the rise of pressure from fan inlet to fan outlet.

Velocity pressure is the pressure corresponding to the average velocity, determined from the volume of airflow at the fan outlet area.

Static pressure is the total pressure diminished by the fan's velocity pressure. Static pressure is a measure of the fan's performance and is reported by the fan manufacturer in their technical literature. Static pressure is also a measure of the resistance to the flow of air thru the heat exchanger.

Power output is expressed in horsepower (or, kilowatts) and is based on fan volume and fan total pressure.

Power input is expressed in horsepower (or, kilowatts) and is the measured power delivered to the fan shaft.

Mechanical efficiency of a fan is the ratio of power output to power input.

Static efficiency of a fan is the mechanical efficiency multiplied by the ratio of static pressure to the total pressure.

The theoretical power required to move a quantity of air may be determined by the following formula:

SI system

$$\text{Theoretical power} = \frac{\left(\frac{m^3}{sec}\right) \cdot \left(\text{Total pressure, [Pa]}, \left[\frac{N}{m^2}\right]\right)}{(1.0)} \quad \text{[watts]}$$

English system

$$\text{Theoretical hp} = \frac{\left(\frac{ft^3}{min}\right) \cdot \left(\text{Total pressure, [in } H_2 0]\right)}{(6356)} \quad \text{[hp]}$$

Pressure and power both vary with air density.

Fan efficiencies may be determined by the following formulae:

Appendix A-Fan Performance

SI system

$$\text{Mechanical (total) efficiency} = \frac{\left(\frac{m^3}{sec}\right) \cdot \left(\text{Total pressure, [Pa]},\left[\frac{N}{m^2}\right]\right)}{(\text{Power input, [watts]})}$$

English system

$$\text{Mechanical (total) efficiency} = \frac{\left[\frac{\left(\frac{ft^3}{min}\right) \cdot (\text{Total pressure, [in } H_20])}{(6356.0)}\right]}{(\text{Horsepower input})}$$

Mechanical efficiency, based on total pressure is applicable for fans operating with high outlet velocity pressure relative to the static pressure, typical of centrifugal fans.

SI system

$$\text{Static efficiency} = \frac{\left(\frac{m^3}{sec}\right) \cdot \left(\text{Static pressure, [Pa]},\left[\frac{N}{m^2}\right]\right)}{(\text{Power input, [watts]})}$$

English system

$$\text{Static efficiency} = \frac{\left[\frac{\left(\frac{ft^3}{min}\right) \cdot (\text{Static pressure, [in } H_20])}{(6356.0)}\right]}{(\text{Horsepower input})}$$

Static efficiency, based on static pressure is applicable to fans with high static pressure relative to the velocity pressure. Static pressure and static efficiency are used more often than mechanical efficiency and total pressure. When a fan operates against no resistance, the static efficiency becomes zero and is meaningless.

Axial Thrust

Total pressure = static pressure + velocity pressure

Velocity pressure is proportional to air velocity.

SI system

$$\text{Static efficiency} = \frac{\left(\frac{m^3}{sec}\right) \cdot \left(\text{Static pressure, [Pa]},\left[\frac{N}{m^2}\right]\right)}{(\text{Power input, [watts]})}$$

English system

$$\text{Static efficiency} = \frac{\left[\frac{\left(\frac{ft^3}{min}\right) \cdot (\text{Static pressure, [in } H_20])}{(6356.0)}\right]}{(\text{Horsepower input})}$$

Appendix A-Fan Performance

SI system

$$\text{Velocity pressure} = \frac{(V^2 \cdot \rho)}{2},\ \left[\text{Pa}, \frac{N}{m^2}\right]$$

$$V = \left[\frac{m}{sec}\right]$$

$$\rho = \text{Density of air}, \left[\frac{Kg}{m^3}\right]$$

English system

$$\text{Velocity pressure} = \left(\frac{V}{4005}\right)^2,\ [\text{in } H_2 0]$$

$$V = \left[\frac{ft}{min}\right]$$

Four inches H_2O = 1000 Pa

SI system

$$\text{Axial thrust} = \left[\frac{\pi \cdot \text{Total pressure} \cdot (\text{Fan diameter})^2}{4}\right], [N]$$

$$\text{Total pressure},\ \left[\text{Pa}, \frac{N}{m^2}\right]$$

Fan diameter, [m]

English system

$$\text{Axial thrust} = \left[\frac{\pi \cdot \text{Total pressure} \cdot (\text{Fan diameter})^2}{(27.68) \cdot 4}\right], [lbf]$$

Total pressure, [in $H_2 0$]

Fan diameter, [inches]

Fan Laws

The performance characteristics of fans of all types follow certain laws, which are useful in predicting the effect upon performance of certain changes in the conditions of operation, or the size of the equipment due to limitations of space, power, and/or speed. In the following categories, Q = air flow, and Pres. = static, velocity, or total pressure. The categories pertaining to fan size apply only to fans, which are geometrically similar, i.e., those in which all of the dimensions are proportional to some linear dimension identified as *size*.

1. Variation in fan speed:

Constant air density – Constant system

a. Q: Varies as fan speed

b. Pres: Varies as square of fan speed

c. Power: Varies as cube of fan speed

2. Variation in fan size:

Constant tip speed – Constant air density

Constant fan proportions – Fixed point of rating

Appendix A-Fan Performance

- **a.** Q: Varies as square of fan diameter
- **b.** Pres: Remains constant
- **c.** rpm: Varies inversely as fan diameter
- **d.** Power: Varies as square of fan diameter

3. Variation in fan size:

At constant rpm – Constant air density

Constant fan proportions – fixed Point of rating

- **a.** Q: Varies as cube of fan diameter
- **b.** Pres: Varies as square of fan diameter
- **c.** Tip speed: Varies as fan diameter
- **d.** Power Varies as fifth power of diameter

4. Variation in air density: (Refer to Appendix F)

Constant volume – Constant system

Fixed fan size – Constant fan speed

- **a.** Q: Constant
- **b.** Pres: Varies as density (SI), specific weight (English)
- **c.** Power: Varies as density (SI), specific weight (English)

5. Variation in air density: (Refer to Appendix F)

Constant pressure – Constant system

Fixed fan size – Variable fan speed

- **a.** Q: Varies inversely as square root of density (SI), specific weight (English)
- **b.** Pres: Constant
- **c.** rpm: Varies inversely as square root of density (SI), specific weight (English)
- **d.** Power: Varies inversely as square root of density (SI), specific weight (English)

6. Variation in air density: (Refer to Appendix F)

Constant weight of air – Constant system

Fixed fan size – Variable fan speed

- **a.** Q: Varies inversely as density (SI), specific weight (English)
- **b.** Pres: Varies inversely as density (SI), specific weight (English)
- **c.** rpm: Varies inversely as density (SI), specific weight (English)
- **d.** Power: Varies inversely as square of density (SI), specific weight (English)

Fan laws at a glance

	Variation in fan speed	**Variation in fan size**		**Variation in air density**		
Fan laws	**Fan law #1**	**Fan law #2**	**Fan law #3**	**Fan law #4**	**Fan law #5**	**Fan law #6**
Parameters	Constant air density & constant system	Constant tip speed & constant air density	Constant rpm & constant air density	Constant volume & constant system	Constant pressure & constant system	Constant weight of air & constant system
		Constant tip proportions & fixed point of rating	Constant fan proportions & fixed point of rating	Fixed fan size & constant fan speed	Fixed fan size & variable fan speed	Fixed fan size & variable fan speed
Q, Air volume, (flow rate)	Varies as fan speed	Varies as square of fan diameter	Varies as cube of fan diameter	Constant	Varies inversely as square root of density or specific weight	Varies inversely as density or specific weight

Appendix A-Fan Performance

Fan laws at a glance (continued)

	Variation in fan speed	Variation in fan size		Variation in air density		
Fan laws	**Fan law #1**	**Fan law #2**	**Fan law #3**	**Fan law #4**	**Fan law #5**	**Fan law #6**
Pressure	Varies as square of fan speed	Remains constant	Varies as square of fan diameter	Varies as density or specific weight	Constant	Varies inversely as density or specific weight
rpm	Constant	Varies inversely as fan diameter	Constant	Constant	Varies inversely as square root of density or specific weight	Varies inversely as density or specific weight
Power	Varies as cube of fan speed	Varies square of fan diameter	Varies as 5th power of diameter	Varies as density or specific weight Varies inversely as 4th power of diameter	Varies as inversely as square root of density or specific weight	Varies as inversely as square of density or specific weight
Tip Speed	N/A	N/A	Varies as fan diameter	N/A		

Example 1

A fan delivers 12,000 cfm at a static pressure of 1 in. H2O when operating at a speed of 400 rpm and requires an input of 4 hp. If in the same installation, 15,000 cfm are required, what will be the speed, static pressure, and power?

Using fan law 1:

Speed = 400 * {15,000/12,000} = 500 rpm

Static pressure = 1 * $\{500/400\}^2$ = 1.56 in. H_2O

Power = 4 * $\{500/400\}^3$ = 7.81 hp

Example 2

A fan delivers 12,000 cfm at 70 °F and normal barometric pressure (density = 0.075 lb per cubic foot) at a static pressure of 1 in. H2O when operating at 400 rpm, and requires 4 hp. If the air temperature is increased to 200 °F (density = 0.0602 lb per cubic foot) and the speed of the fan remains the same, what will be the static pressure and power?

Using fan law 4:

Static pressure = 1 * {0.0602/0.075} = 0.80 in. H_2O

Power = 4 * {0.0602/0.075} = 3.20 hp

Example 3

If the speed of the fan in example 2 is increased to produce a static pressure of 1 in. H2O at 200 °F, as at 70 °F, what will be the speed, capacity, and static power?

Using fan law 5:

Speed = 400 * SQRT {0.0705/0.0602} = 446 rpm

Capacity = 12,000 * SQRT {0.075/0.0602} = 13,392 cfm (measured at 200°F)

Power = 4 * SQRT {0.075/0.0602} = 4.46 hp

Appendix A-Fan Performance

Example 4

If the speed of the fan in the previous examples is increased to deliver the same weight of air (same cooling capacity) at 200°F as at 70°F, what will be the speed, capacity, static pressure, and power?

Heat transfer is determined by the mass, or weight, of the air presented to the heat exchanger, or radiator.

Using fan law 6:

Speed = 400 * {0.075/0.0602} = 498 rpm

Capacity = 12,000 * {0.075/0.0602} = 14,945 cfm (measured at 200 F)

Static Pressure = 1 * {0.075/0.0602} = 1.25 in. H2O

Power = 4 * {0.075/0.0602}2 = 6.20 hp

The fan laws may be combined to give other overall values. One useful combination is the product of Laws 1 and 3, which gives the following relationships:

- Capacity (flow rate of air) varies as the ratio of size cubed, times the ratio of the rpm.
- Pressure varies as the ratio of size squared, times the ratio of the rpm squared.
- Power varies as the ratio of the size to the fifth power, times the ratio of the rpm cubed.

Centrifugal fans produce pressure from two independent sources: from the centrifugal force created by rotating the enclosed air column, and from the kinetic energy imparted to the air by virtue of its velocity leaving the impeller. The energy imparted to the air depends on the velocities and is dependent on the curvature of the fan blades. Therefore, for fans with forward curved blades, the energy per pound of air rises rapidly with an increase of air delivery. For fans with backward curved blades, the energy per pound of air may decrease with air delivery (flow rate). For fans with straight blades, the energy per pound of air is roughly constant, regardless of air delivery (flow rate). A unique characteristic of centrifugal fans is that the maximum power required by the fan is found at maximum delivery. Or, otherwise stated, the minimum power required by the fan is found at zero delivery, or under stall conditions.

Axial-flow fans develop none of their static pressure by centrifugal force; all of the static pressure is derived from the change in velocity in passing thru the impeller and its conversion into static pressure. They are inherently high velocity fans and are very dependent on the shape of the blade. Since any particular shape of blade can only be correct for a narrow range of capacity at constant speed, the performance curves for individual fan blade shapes are unique and vary significantly from manufacturer, to manufacturer. To absorb energy, the air must be given a tangential motion in passing thru the impeller; as with the centrifugal fan, the pressure rises generally from free delivery to no delivery, but may drop significantly when the capacity decreases below a certain volume. The drop in pressure is indicative that a stall condition occurs and the blades cease to function in the normal manner.

Fan manufacturers generally agree that the tip clearance around the fan's blades is significant and will contribute to the performance, delivery, and operating efficiency of the fan. Likewise, the type of shroud that surrounds the fan, the axial position of the fan within the shroud, and the clearance between the leading edge of the fan and the cooler can contribute significantly to the performance, operating efficiency, and noise of operation of the fan. The system designer is advised to consult with both the fan and cooler manufacturer for these specific design elements for the system under consideration.

Appendix B-Fan Drive Sizing Equations

Fan Drive Sizing Equations and Derivations

Fan drive sizing equations - SI system

Pf = Rated fan power, [kW]
Nr = Rated fan speed, [rpm]
Nfr = Fan speed, [rpm]
Cf = Fan power coefficient, $\left[\frac{kW}{rpm^3}\right]$

1. $Pf \equiv Cf \cdot Nfr^3 \equiv \frac{Tfr \cdot Nfr}{9549}$, [kW]

Tfr = Rated fan torque, [N•m]

2. $Tfr = \frac{\Delta P \cdot Dm \cdot n_{tm}}{20\pi}$, [N•m]

ΔP = Pressure drop across hydraulic motor, [bar]

$Dm \equiv$ Motor displacement, $\left[\frac{cm^3}{rev}\right]$

Dm = (Motor displacement, $\left(\left[\frac{in^3}{rev}\right] \cdot (2.54\ \frac{cm}{in})^3\right)$, $\left[\frac{cm^3}{rev}\right]$

n_{tm} = Motor mechanical efficiency, no dimension

n_{vm} = Motor volumetric efficiency, no dimension

n_{vp} = Pump volumetric Efficiency, no dimension

3. $Pf = \frac{(\Delta P \cdot Nf \cdot Dm \cdot \eta tm)}{(20\pi \cdot 9549)}$, [kW]

4. $Qm = \frac{Nf \cdot Dm}{(1000.0 \cdot \eta vm)}$, $\left[\frac{L}{min}\right]$

5. $Nf \cdot Dm = (Cf \cdot Nf^3) \cdot \frac{(20\pi \cdot 9549.0)}{(\Delta P \cdot \eta tm)}$, $\left[\frac{cm^3}{min}\right]$

6. $Qm = (Cf \cdot Nf^3) \cdot \frac{(20\pi \cdot 9549.0)}{(\Delta P \cdot \eta tm \cdot 1000 \cdot \eta vm)}$, $\left[\frac{L}{min}\right]$

Re-arranging #5. above,

7. $Nf = \sqrt{\frac{\Delta P \cdot Dm \cdot \eta tm}{(600{,}000 \cdot Cf)}}$, [rpm]

And also,

8. $Qm = \left[\frac{\sqrt{\frac{\Delta P \cdot Dm^3 \cdot \eta tm}{(600{,}000 \cdot Cf)}}}{(1{,}000 \cdot \eta vm)}\right]$, $\left[\frac{L}{min}\right]$

Likewise,

9. $Dm = \left[\frac{Nf^2 \cdot 600{,}000 \cdot Cf}{\Delta P \cdot n_{tm}}\right]$, $\left[\frac{cm^3}{rev}\right]$

and,

10. $\Delta P = \left[\frac{Cf \cdot Nf^2 \cdot 600{,}000}{Dm \cdot \eta_{tm}}\right]$, [bar]

Appendix B-Fan Drive Sizing Equations

For charge pump sizing equations, refer to the product specific Technical Information manuals, and Systems Applications manuals.

Fan drive sizing equations - English system

Pf = Rated fan power, [hp]
Nfr = Rated fan speed, [rpm]
Nf = Fan speed, [rpm]
Cf = Fan power coefficient, $\left[\frac{hp}{rpm^3}\right]$

1. $Pf \equiv Cf \cdot Nfr^3 \equiv \frac{Tfr \cdot Nfr}{63{,}025}$, [hp]

Tfr = Rated fan torque, [lbf • in]

2. $Tfr = \frac{\Delta P \cdot Dm \cdot \eta_{tm}}{2\pi}$, [lbf•in]

ΔP = Pressure drop across hydraulic motor, [psid]

Dm = Motor displacement, $\left[\frac{in^3}{rev}\right]$

$$Dm \equiv \frac{(\text{Motor Displacement}, \left[\frac{cm^3}{rev}\right])}{(2.54\left[\frac{cm}{in}\right])^3}, \left[\frac{in^3}{rev}\right]$$

η_{tm} = Motor mechanical efficiency, no dimension

η_{vm} = Motor volumetric efficiency, no dimension

η_{vp} = Pump volumetric efficiency, no dimension

3. $Pf = \frac{(\Delta P \cdot Nf \cdot Dm \cdot \eta tm)}{(2\pi \cdot 63025)}$, [hp]

4. $Qm = \frac{Nf \cdot Dm}{(231 \cdot \eta vm)}$, [US gal/min]

5. $Nf \cdot Dm = (Cf \cdot Nf^3) \cdot \frac{(2\pi \cdot 63{,}025)}{(\Delta P \cdot \eta tm)}$, $\left[\frac{in^3}{min}\right]$

6. $Qm = (Cf \cdot Nf^3) \cdot \frac{(2\pi \cdot 63{,}025)}{(\Delta P \cdot \eta tm \cdot 231 \cdot \eta vm)}$, [US gal/min]

Re-arranging #5. above,

7. $Nf = \sqrt{\frac{\Delta P \cdot Dm \cdot \eta tm}{(396{,}000 \cdot Cf)}}$, [rpm]

And also,

8. $Qm = \left[\frac{\sqrt{\frac{\Delta P \cdot Dm^3 \cdot \eta tm}{(396{,}000 \cdot Cf)}}}{(231 \cdot \eta vw)}\right]$, [US gal/min]

Likewise,

9. $Dm = \left[\frac{Nf^2 \cdot 396{,}000 \cdot Cf}{\Delta P \cdot \eta_{tm}}\right]$, $\left[\frac{in^3}{rev}\right]$

and,

10. $\Delta P = \left[\frac{Cf \cdot Nf^2 \cdot 396{,}000}{Dm \cdot \eta_{tm}}\right]$, [psid]

Appendix B-Fan Drive Sizing Equations

For charge pump sizing equations, refer to the product specific Technical Information manuals, and Systems Applications manuals.

Therefore, comparing equations 7 thru 10 on the previous pages, for any given combination of fan configuration and motor displacement, there is a unique relationship between system pressure, motor flow, and motor speed.

Hydraulic System Comparisons

One application of the sizing equations is to use them to relate the fan speed/pressure relationship from one condition to another.

SI system

From equation 7,

$$Nf^2 = \left[\frac{\Delta P \cdot Dm \cdot \eta_{tm}}{(600{,}000 \cdot Cf)}\right]$$

Therefore at condition 1:

$$Nf_1^2 = \left[\frac{\Delta P_1 \cdot Dm \cdot \eta_{tm}}{(600{,}000 \cdot Cf)}\right]$$

and at condition 2:

$$Nf_2^2 = \left[\frac{\Delta P_2 \cdot Dm \cdot \eta_{tm}}{(600{,}000 \cdot Cf)}\right]$$

Then,

$$\frac{Nf_1^2}{Nf_2^2} = \frac{\Delta P_1}{\Delta P_2}$$

$$Nf_2 = Nf_1 \cdot \sqrt{\frac{\Delta P_2}{\Delta P_1}}$$

English system

From equation 7,

$$Nf^2 = \left[\frac{\Delta P \cdot Dm \cdot \eta_{tm}}{(396{,}000 \cdot Cf)}\right]$$

Therefore at condition 1:

$$Nf_1^2 = \left[\frac{\Delta P_1 \cdot Dm \cdot \eta_{tm}}{(396{,}000 \cdot Cf)}\right]$$

and at condition 2:

$$Nf_2^2 = \left[\frac{\Delta P_2 \cdot Dm \cdot \eta_{tm}}{(396{,}000 \cdot Cf)}\right]$$

Then,

$$\frac{Nf_1^2}{Nf_2^2} = \frac{\Delta P_1}{\Delta P_2}$$

$$Nf_2 = Nf_1 \cdot \sqrt{\frac{\Delta P_2}{\Delta P_1}}$$

If the theoretical trim pressure is 2822 psid at a fan speed of 2000 rpm, what is the fan speed at a minimum standby pressure of 310 psid?

$$Nf_2 = Nf_1 \cdot \sqrt{\frac{\Delta P_2}{\Delta P_1}}$$

$$Nf_2 = 2000 \cdot \sqrt{\frac{310}{2822}} \quad = 663 \text{ rpm}$$

If the minimum standby pressure suddenly changes from 310 psid, to 400 psid, due to the external circuit, what is the resulting fan speed?

$$Nf_2 = Nf_1 \cdot \sqrt{\frac{\Delta P_2}{\Delta P_1}}$$

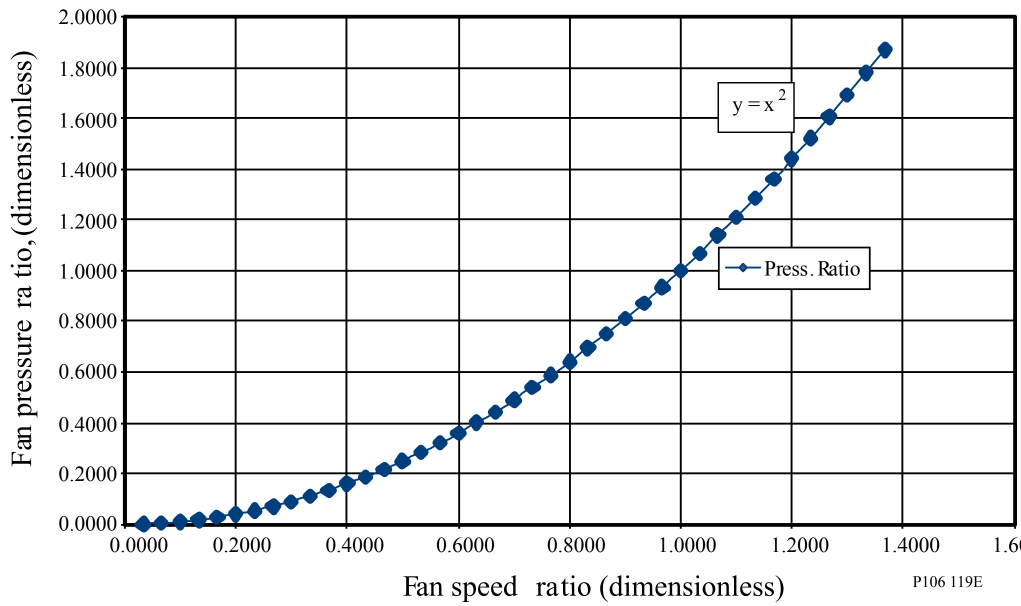
$$Nf_2 = 663 \cdot \sqrt{\frac{400}{310}} \quad = 753 \text{ rpm}$$

The graph below illustrates this quadratic relationship between fan system delta pressure and fan speed. The graph is normalized by dividing the individual operating parameters of pressure and speed by the trim pressure and trim speed of the fan so that the curve is representative of all fan systems.

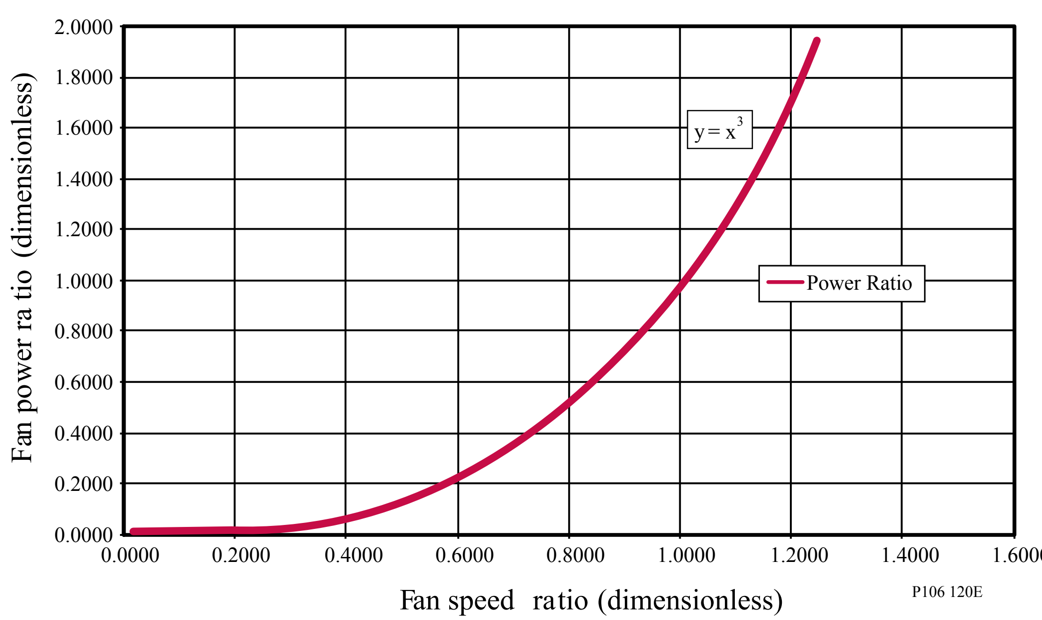
Appendix B-Fan Drive Sizing Equations

Consequently, the curve converges on the coordinates of (1.000, 1.000) for both the fan pressure ratio and the fan speed ratio.

Pressure ratio vs fan speed ratio



Power ratio vs fan speed ratio



On examination of the curve, one can see:

- To increase fan speed 10% above the trim speed we would need to increase the delta pressure across the motor by 21%
- To reduce fan speed below 32% of the trim speed we would need to reduce the delta pressure across the motor by more than 90%.

Appendix C-Fan Drive Sizing Equations, using Variable Displacement Motors

Hydraulic Systems with 2 Position, Variable Displacement Motors, Equations and Derivations

Some systems may require additional cooling when the engine is at the low idle speed. A system with a fixed displacement motor may require additional pump flow to accomplish this. If the pump flow is limited, there is the possibility of providing additional cooling by utilizing a variable displacement motor in the hydraulic circuit. For normal operation, the variable displacement motor is held in the maximum displacement position in order to minimize the operating pressure for any given cooling requirement. But, when the engine is at the low idle speed, the control system can command the motor to its minimum displacement position to take full advantage of the flow that is available from the pump. In both conditions, the fan speed is modulated in response to the temperature of the coolant to satisfy the cooling needs of the system.

If you would like to determine the optimum minimum displacement for a variable motor, which would provide the maximum fan speed when, the engine speed is at low idle and the system pressure is at the trim pressure setting:

The following analysis assumes that the bypass flow around the motor is zero.

SI system

SI system

Assume Qm = Qp,

$$\frac{Nf \cdot Dm}{\eta_{vm}} = Np \cdot Dp \cdot \eta_{vp}$$

Therefore:

$$11.\ Nf = \frac{Np \cdot Dp \cdot \eta_{vp} \cdot \eta_{vm}}{Dm} = \left[\frac{(Ne \cdot R) \cdot Dp\ \eta_{vp} \cdot \eta_{vm}}{Dm}\right]$$

Ne = Engine speed, rpm (In this case, Ne = engine low idle speed.)
R = Engine-pump gear ratio, no dimension.
Combining eq. 1. and eq. 11.,

$$12.\ Pf = Cf \cdot Nf^3 = \frac{\Delta P \cdot Dm \cdot \eta_{tm}}{600{,}000} \cdot \frac{Ne \cdot R \cdot Dp\ \eta_{vp} \cdot \eta_{vm}}{Dm},\ [kW]$$

$$13.\ Nf = \left[\frac{\Delta P \cdot Ne \cdot R \cdot Dp\ \eta_{tm} \cdot \eta_{vp} \cdot \eta_{vm}}{600{,}000 \cdot Cf}\right]^{1/3},\ [rpm]$$

Combining equation 11 and equation 13,

$$14.\ \left[\frac{Ne \cdot R \cdot Dp\ \eta_{vp} \cdot \eta_{vm}}{Dm}\right]^3 = \left[\frac{\Delta P \cdot Ne \cdot R \cdot Dp\ \eta_{tm} \cdot \eta_{vp} \cdot \eta_{vm}}{600{,}000 \cdot Cf}\right]$$

Solving for Dm,

$$15.\ Dm = \left[\frac{600{,}000 \cdot Cf}{\Delta P\ \eta_{tm}} \cdot (Ne \cdot R \cdot Dp\ \eta_{vp} \cdot \eta_{vm})^2\right]^{1/3},\quad \left[\frac{in^3}{rev}\right]$$

Appendix C-Fan Drive Sizing Equations, using Variable Displacement Motors

English system

English system

Assume Qm = Qp,

$$\frac{Nf \cdot Dm}{\eta_{vm}} = Np \cdot Dp \cdot \eta_{vp}$$

Therefore:

$$11.\ Nf = \left[\frac{Nf \cdot Dp \cdot \eta_{vp} \cdot \eta_{vm}}{Dm}\right] = \left[\frac{(Ne \cdot R) \cdot Dp\, \eta_{vp} \cdot \eta_{vm}}{Dm}\right]$$

Ne = Engine speed, rpm (In this case, Ne = engine low idle speed.)
R = Engine/pump gear ratio, no dimension.
Combining eq. 1. and eq. 11.,

$$12.\ Pf = Cf \cdot Nf^3 = \frac{\Delta P \cdot Dm \cdot \eta_{tm}}{396{,}000} \cdot \frac{Ne \cdot R \cdot Dp\, \eta_{vp} \cdot \eta_{vm}}{Dm},\ [hp]$$

$$13.\ Nf = \left[\frac{\Delta P \cdot Ne \cdot R \cdot Dp\, \eta_{tm} \cdot \eta_{vp} \cdot \eta_{vm}}{396{,}000 \cdot Cf}\right]^{1/3},\ [rpm]$$

Combining eq. 11. and eq. 13.,

$$14.\ \left[\frac{Ne \cdot R \cdot Dp\, \eta_{vp} \cdot \eta_{vm}}{Dm}\right]^3 = \left[\frac{\Delta P \cdot Ne \cdot R \cdot Dp\, \eta_{tm} \cdot \eta_{vp} \cdot \eta_{vm}}{396{,}000 \cdot Cf}\right]$$

Solving for Dm,

$$15.\ Dm = \left[\frac{396{,}000 \cdot Cf}{\Delta P \cdot \eta_{tm}} \cdot (Ne \cdot R \cdot Dp\, \eta_{vp} \cdot \eta_{vm})^2\right]^{1/3},\quad \left[\frac{in^3}{rev}\right]$$

This is the optimum minimum motor displacement at which the fan speed will be maximum for the given pump flow at engine idle and at the system's trim pressure. An ExcelTM spreadsheet has been developed to calculate the optimum minimum displacement for the variable displacement motor. Consult your Danfoss representative for assistance.

Spreadsheet to Calculate the Optimum Minimum Displacement for 2 Position Variable Motor

Spreadsheet calculator tool (example)

Rated fan power, hp	40	Input data *only* in second column:		
Rated fan speed, rpm	2400	Optimum displ. Dm =	In^3/rev 1.230	cc/rev 20.15
Desired fan trim speed, rpm	2600			
Trim pressure, Psig	3205			
Motor displ., cc/rev	44			
Motor torque eff	90%			
Cf = hp_fan/Nf^3	2.89E-09			
Engine idle speed, rpm	700			
Engine: pump ratio, _:1	1			
Pump displ, cc/rev	55			
Pump displ, In^3/rev	3.36			

Appendix C-Fan Drive Sizing Equations, using Variable Displacement Motors

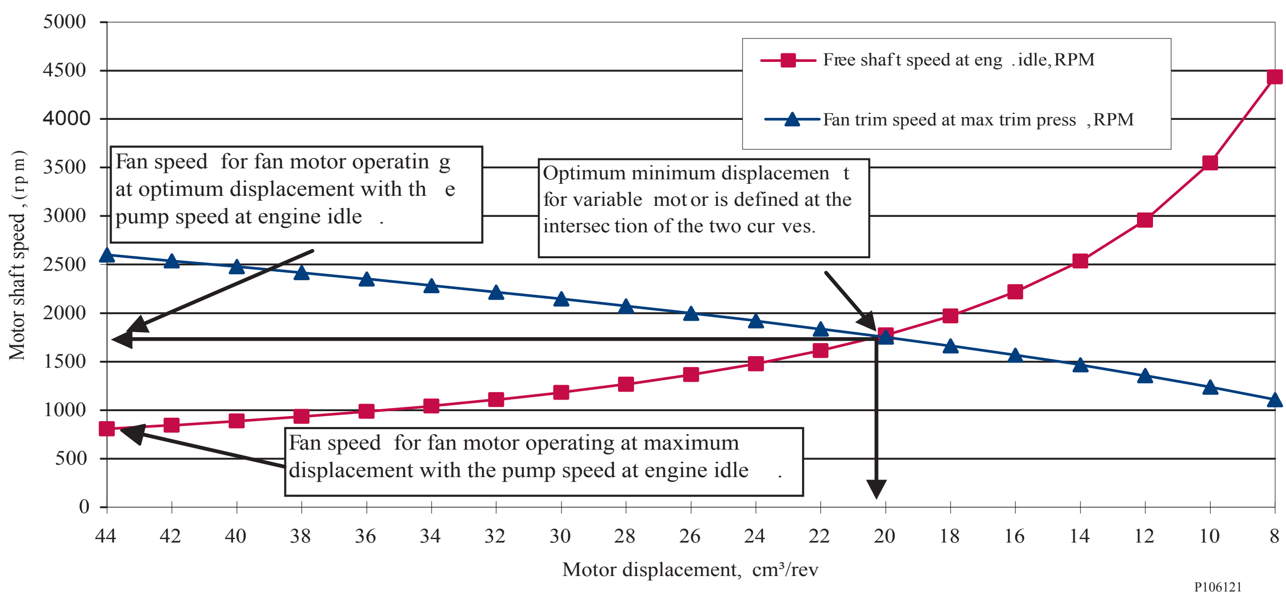
Spreadsheet calculator tool (example) (continued)

Motor vol eff	96%			
Pump vol eff	96%			

This relationship is illustrated in the graph below.

- The curve described by the square data points represents the speed of the motor shaft as a function of motor displacement when the pump's displacement is constant, and the pump's input speed is at the engine idle speed.
- The curve described by the triangular data points represents the speed of the motor shaft as a function of the fan power coefficient, the trim pressure and the motor's displacement.
- For the system parameters that have been chosen, the optimum minimum motor displacement for the variable motor will be found at the intersection of these two curves.

Variable motor shaft speed vs. displacement

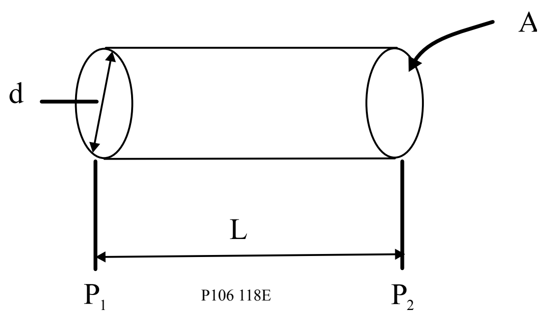


Fan rated power = 40 hp
Fan rated speed = 2400 rpm

Appendix D-Pressure change due to transient flow in a passage

Pressure Change due to Transient Flow in a Passage, Equations and Derivations

Flow passage



ΔP = (P1 – P2), psid	Required pressure to achieve fluid acceleration, a
a , in/sec²	Fluid acceleration in L length of conduit
L = 1ft	Length of conduit
A, in²	Area of conduit
m, lbf-sec²/in⁴	Fluid mass within the L length of conduit
S.G. = 0.861	Specific gravity of the fluid
Q =V • A, in³/sec	Flow rate (in³/sec = GPM x 3.85)
V, ft/sec	Velocity of fluid in conduit
D, in	Diameter of the conduit
F, lbf	Force
ρ	Mass density of the fluid
g	Acceleration of gravity
γ	Weight density of the fluid
γ/g = ρ	Weight density of fluid/acceleration of gravity = mass density of the fluid
t	Time interval, seconds
D	Pump displacement

$$F = \Delta P \cdot A, \text{lbf} = m \cdot a = \rho_f \cdot Q \cdot V,$$

$$\Delta P \cdot A = \rho_f \cdot Q \cdot V = \rho \cdot \left(\frac{L \cdot A}{t}\right) \cdot V$$

$$\Delta P \cdot A = \rho_f \cdot \left(\frac{L \cdot A}{t}\right) \cdot \frac{Q}{A}$$

$$\Delta P = \rho_f \cdot \left(\frac{L \cdot Q}{t \cdot A}\right)$$

and, $\rho_f = S.G \cdot \rho_{H20}$

Therefore:

$$\Delta P = \frac{(1.43\ E\text{-}3)(S.G.)(L)(Q)}{t \cdot d^2}, \text{Psid}$$

Appendix D-Pressure change due to transient flow in a passage

SI system

$\frac{\Delta P}{L} = (2.122E-7) \cdot \left(\frac{S.G \cdot Q}{t \cdot d^2}\right), \left[\frac{bar}{meter}\right]$

Q = flow, $\left[\frac{L}{min}\right]$

L = length, [m]

d = diameter, [m]

and,

$\frac{\Delta P}{L} = (0.2122) \cdot \left(\frac{S.G \cdot Q}{t \cdot d^2}\right), \left[\frac{bar}{meter}\right]$

when:

d = diameter, [mm]

also,

$\frac{\Delta P}{L} = (2.122E-4) \cdot \left(\frac{S.G \cdot N \cdot D}{t \cdot d^2}\right), \left[\frac{bar}{meter}\right]$

when:

d = diameter, [mm]

D = displacement, $\left[\frac{cm^3}{Rev}\right]$

N = speed, [rpm]

and,

$\frac{\Delta P}{L} = (2.122E-4) \cdot \left(\frac{S.G \cdot D \cdot a}{d^2}\right), \left[\frac{bar}{meter}\right]$

when:

d = diameter, [mm]

D = displacement, $\left[\frac{cm^3}{Rev}\right]$

$a = \frac{\Delta N}{t}$, average acceleration, $\left[\frac{rpm}{sec}\right]$

English system

$\frac{\Delta P}{L} = (5.4996E-3) \cdot \left(\frac{S.G \cdot Q}{t \cdot d^2}\right), \left[\frac{psid}{ft}\right]$

Q = flow, [GPM]

L = length, [ft]

d = diameter, [in]

$\frac{\Delta P}{L} = (3.7189E-3) \cdot \left(\frac{Q}{t}\right), \left[\frac{psid}{ft}\right]$

when:

$S.G = 0.861$

$A = 1\ in^2$

$\frac{\Delta P}{L} = (2.3808E-3) \cdot \left(\frac{S.G \cdot D}{d^2}\right) \cdot a, \left[\frac{psid}{meter}\right]$

when:

$a = \frac{\Delta N}{t}$ = average acceleration, $\left[\frac{rpm}{sec}\right]$

D = displacement, $\left[\frac{in^3}{rev}\right]$

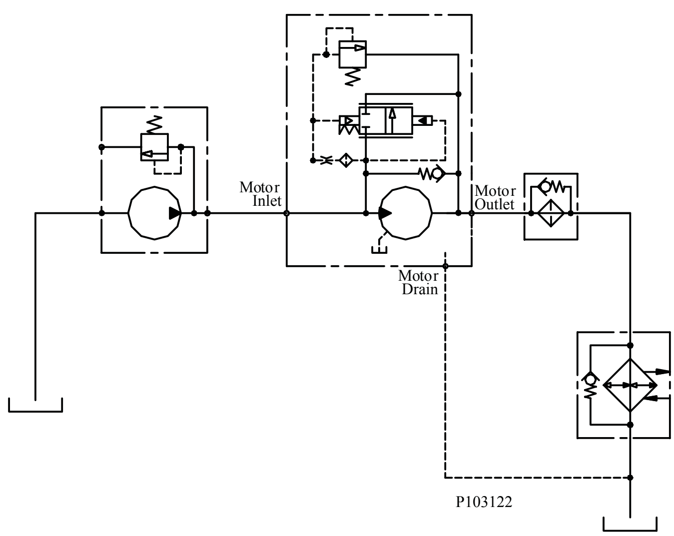
Appendix E-Influence of Bypass Valve Pressure Drop in Open Circuit Systems

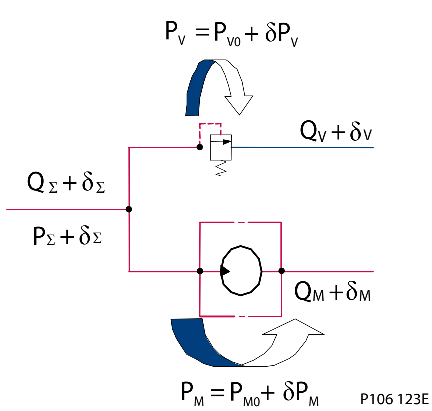
In applications where the system is sized close to the maximum limits of fan speed, motor speed, or component pressure ratings, it may be necessary to determine the influence of the bypass valve's pressure-flow characteristics on the system. In a typical circuit, there will be a rise in pressure across a component with an increase in flow. This is normal, and the system's designer is encouraged to check whether this characteristic will have an adverse effect on the system and its components.

In this section, we will determine the effect of the pressure rise with flow characteristic of the bypass valve on the change in trim speed of the fan and the trim pressure across the hydraulic motor. In most pressure regulating valves there is a range of flows where the pressure rise is either linear, or nearly linear, with increases in flow. This is illustrated below for an electro-hydraulic proportional pressure control. Adjacent to each of the characteristic curves is a coefficient, which is the nominal slope of each curve. With this coefficient and other system parameters that are described below: the system designer can determine the change in trim speed and trim pressure with increases in engine speed above the engine set point that has been selected.

For the purposes of this derivation, the basic modulating fan drive circuit may be simplified as shown below:

Basic modulating fan drive circuit





As can be seen in the simplified schematic, all of the pump flow, QS goes thru either the bypass valve, Q_V, or the fan motor, Q_M. At the initial condition for our derivation, the flow thru the bypass valve is zero and all of the pump flow is passing thru the motor. This is the *trim point*, when the engine is at the engine set point. Above the trim point, a portion of the flow will pass thru the motor and the remaining flow will pass thru the bypass valve. The proportion of the flow going thru each will be determined by the pressure-flow characteristics of the valve and the pressure-flow relationship of the motor-fan that is determined by the fan laws.

Appendix E-Influence of Bypass Valve Pressure Drop in Open Circuit Systems

$$Q_{\Sigma 0} = Q_{V0} + Q_{M0}$$

$$Q_{\Sigma 0} + \delta Q_\Sigma = Q_{V0} + \delta Q_V + Q_{M0} + \delta Q_M$$

$$@t = 0,\ Q_{V0} = 0.0$$

$$\therefore Q_{\Sigma 0} = Q_{M0}$$

$$\therefore \delta Q_\Sigma = \delta Q_V + \delta Q_M$$

From the graph of ΔPv vs ΔQv,

$$P_V = P_0 + Kp \cdot \delta Q_V$$

$$Kp \equiv \left(\frac{\Delta Pv}{\Delta Qv}\right),$$

From the fan laws, $\left(\frac{P_2}{P_1}\right) = \left(\frac{Q_2}{Q_1}\right)^2$,

Therefore, $\delta P_m = \left[P_0 \left(\frac{Q_{M0} + \delta Q_M}{Q_{M0}}\right)^2 - P_0\right]$,

$\delta P_m \equiv \delta P_V$, therefore:

$$\delta P_V = P_0 \cdot \left[\left(\frac{Q_{M0} + \delta Q_M}{Q_{M0}}\right)^2 - 1\right.,$$

$$\frac{\delta P_V}{P_0} = \left[\left(\frac{Q_{M0} + \delta Q_M}{Q_{M0}}\right)^2 - 1\right],$$

Expanding the right side of the equation,

$$\frac{\delta P_V}{P_0} = \left[\left(1 + 2\left(\frac{\delta Q_M}{Q_{M0}}\right) + \left(\frac{\delta Q_M}{Q_{M0}}\right)^2\right) - 1\right],$$

If $\left(\frac{\delta Q_M}{Q_{M0}}\right)^2 \ll \left[2\left(\frac{\delta Q_M}{Q_{M0}}\right)\right]$,

This is valid for $\delta Q_M \leq 0.5 \cdot \delta Q_{M0}$

Then, $\frac{\delta P_V}{P_0} = 2\left(\frac{\delta Q_M}{Q_{M0}}\right)$,

Since, $\delta P_V = Kp \cdot \delta Q_V = Kp \cdot (\delta Q_\Sigma - \delta Q_M)$,

Therefore,

$$\frac{Kp}{P_0}(\delta Q_\Sigma - \delta Q_M) = 2\left(\frac{\delta Q_M}{Q_{MO}}\right),$$

Expanding,

$$Kp \cdot \delta Q_\Sigma = P_0 \cdot \left[2\left(\frac{\delta Q_M}{Q_{M0}}\right)\right] + Kp \cdot \delta Q_M,$$

$$Kp \cdot \delta Q_\Sigma = \frac{\delta Q_M}{Q_{M0}} \cdot \{2P_0 + Kp \cdot Q_{M0}\},$$

Then, $\delta Q_M = \left\{\frac{Kp \cdot \delta Q_\Sigma \cdot Q_{M0}}{(2P_0 + Kp \cdot Q_{M0})}\right\}$,

Therefore, $\delta N_M = \frac{\delta Q_M}{D_M}$,

Likewise;

$$(\delta Q_V) = (\delta Q_\Sigma - \delta Q_M),$$

$$\delta Q_V = \delta Q_\Sigma \cdot \left\{1 - \left[\frac{Kp \cdot Q_{M0}}{(2P_0 + Kp \cdot Q_{M0})}\right]\right\},$$

Then,

$$\delta P_V = Kp \cdot \delta Q_\Sigma \cdot \left\{1 - \left[\frac{Kp \cdot Q_{M0}}{(2P_0 + Kp \cdot Q_{M0})}\right]\right\}$$

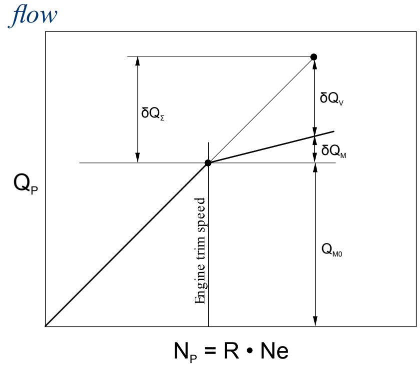
Summarizing then:
If we know the increase in pump flow, δQ_Σ above the engine set point, the flow thru the motor at the engine set point, Q_{M0}, the motor displacement, D_M, the initial trim pressure, P_0, and the bypass valve's flow-pressure coefficient, Kp; then we can determine the increase in fan speed, δN_M, above the theoretical trim speed, N_{M0}.

$$\delta N_M = \left\{\frac{Kp \cdot \delta Q_\Sigma \cdot Q_{M0}}{D_M \cdot (2P_0 + Kp \cdot Q_{M0})}\right\}$$

Likewise, we can determine the change in pressure across the fan motor, δP_V, as shown.

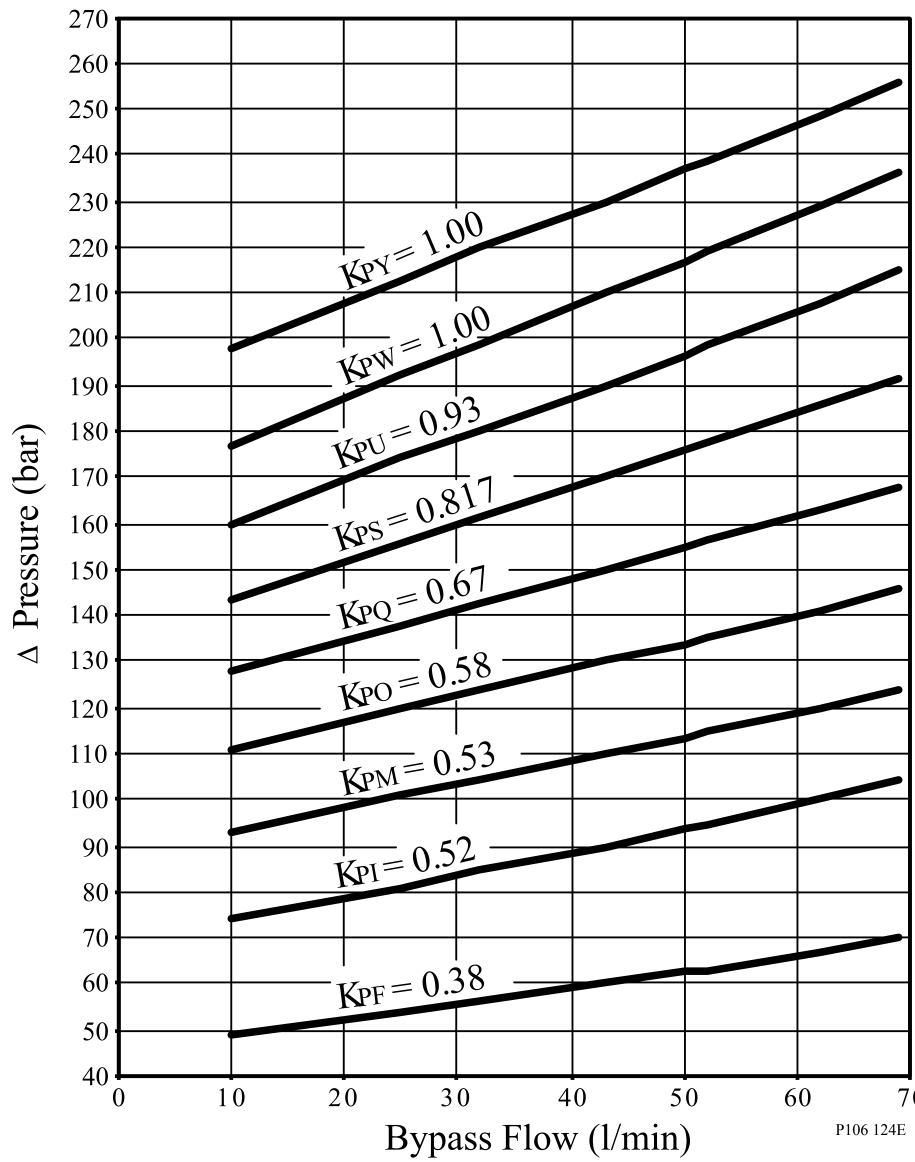
$$\delta P_V = Kp \cdot \delta Q_\Sigma \cdot \left\{1 - \left[\frac{Kp \cdot Q_{M0}}{(2P_0 + Kp \cdot Q_{M0})}\right]\right\}$$

Bypass valve and fan motor flow vs. pump flow



Appendix E-Influence of Bypass Valve Pressure Drop in Open Circuit Systems

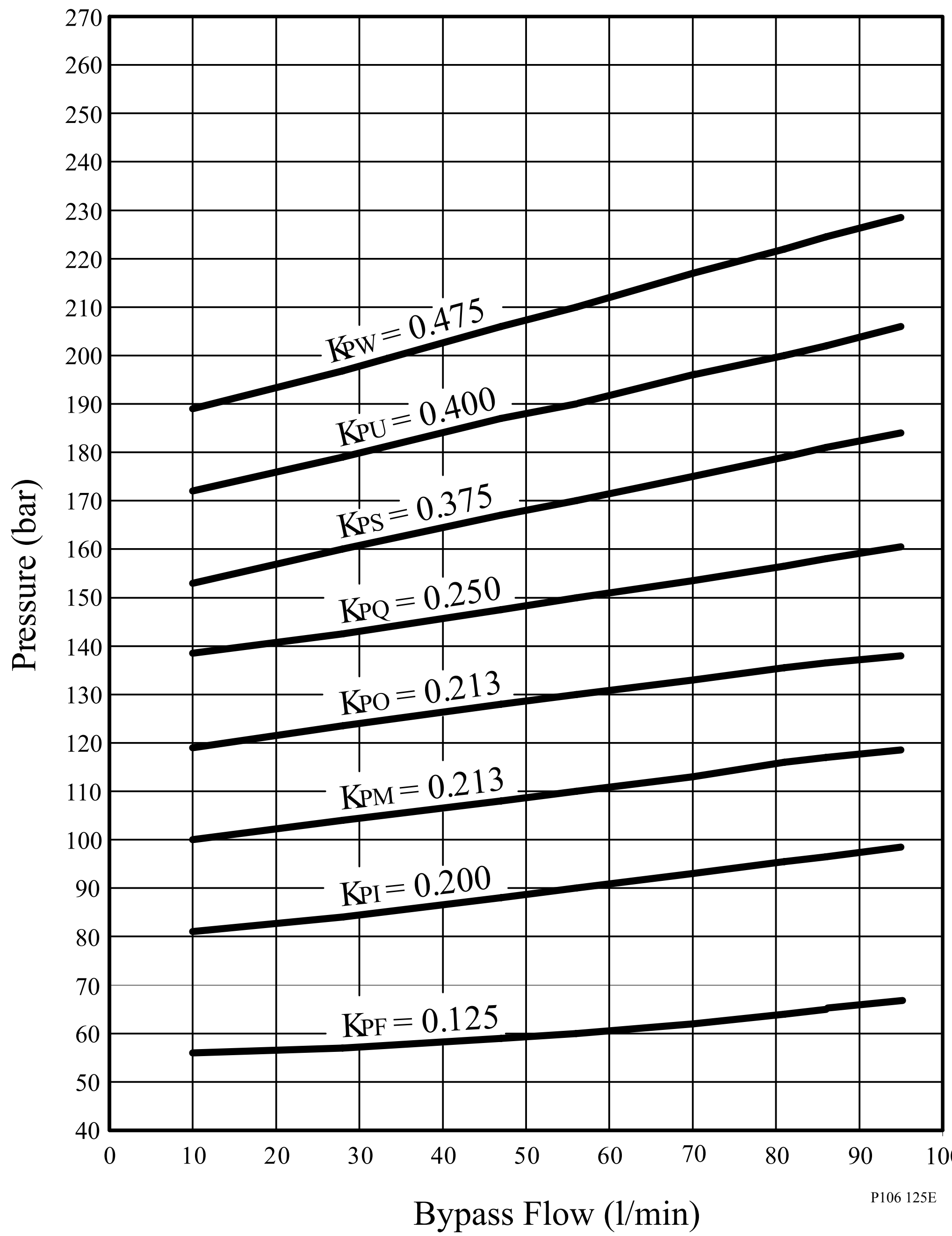
Pressure vs. flow diagram for Comatrol PRV10 valve identification



Y, W, U, etc. - Refer to electro-hydraulic proportional valve nomenclature code in 520L0588 Comatrol Cartridge Valves Technical Information.

Appendix E-Influence of Bypass Valve Pressure Drop in Open Circuit Systems

Pressure vs. flow diagram for Comatrol PRV12 valve identification



W, U, S, etc. - Refer to electro-hydraulic proportional valve nomenclature code in 520L0588 Comatrol Cartridge Valves Technical Information.

Appendix F1-Influence of temperature, pressure and relative humidity on specific weight of air

Influence of Temperature, Pressure and Relative Humidity on Specific Weight of Air

$$v=0.0749 \cdot \left[\frac{530}{460+T}\right] \cdot \left[\frac{P-0.3786 \cdot RH \cdot Pvs}{29.92}\right]$$

Where:

v = Specific Weight of Air, lbf/ft3

P = Standard Barometric Pressure, in Hg

RH = Relative Humidity, (%/100)

Pvs = Saturation Water-Vapor Pressure, in Hg

T = Temperature, °F

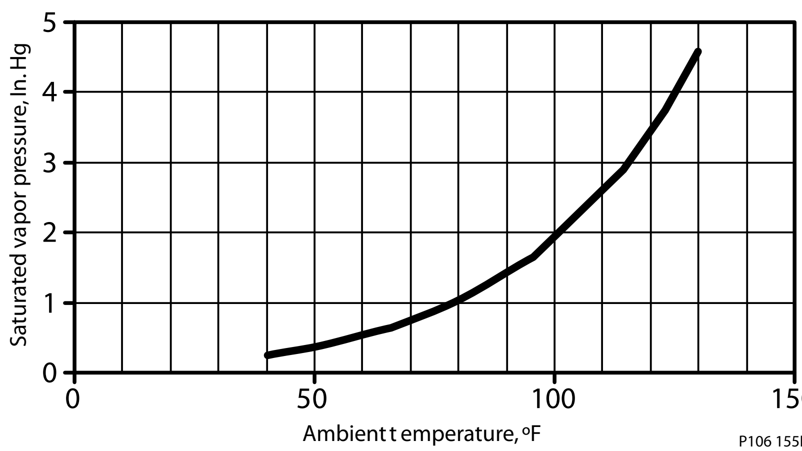
H = Altitude, ft.

Standard barometric pressure varies with altitude

$P = (1.34955E\text{-}8)H^2 - (1.07145E\text{-}3)H + 29.92$, In. Hg

Accurate within +0.0/- 0.5 In. Hg from - 1000 to 15000 ft

Saturated water vapor pressure vs ambient temperature, ºF



$y = 4E\text{-}06x^3 - 0.0005x^2 + 0.0343x - 0.5782$

Temperature, vs pressure (in. Hg)

Temp., ºF	Pvs, in. Hg
40	0.2478
42	0.2671
44	0.2891
46	.03119
48	0.3363
50	0.3624
52	0.3905
54	0.42
56	0.4518
58	0.4854

Appendix F1-Influence of temperature, pressure and relative humidity on specific weight of air

Temp., ºF	Pvs, in. Hg
60	0.5214
62	0.5597
64	0.6004
66	0.6438
68	0.6898
70	0.7387
72	0.7906
74	0.8456
76	0.904
78	0.9657

Temp., ºF	Pvs, in. Hg
80	1.032
82	1.101
84	1.175
86	1.253
88	1.335
90	1.421
92	1.513
94	1.609
96	1.711
98	1.818

Temp., ºF	Pvs, in. Hg
100	1.932
102	2.0528
104	2.1786
106	2.3109
108	2.4502
110	2.5966
112	2.7505
114	2.9123
116	3.0821
118	3.3606

Temp., ºF	Pvs, in. Hg
120	3.4474
122	3.6436
124	3.8492
126	4.0649
128	4.2907
130	4.5272

Appendix F2-Influence of Altitude on Atmospheric Pressure

Influence of Altitude on Atmospheric Pressure

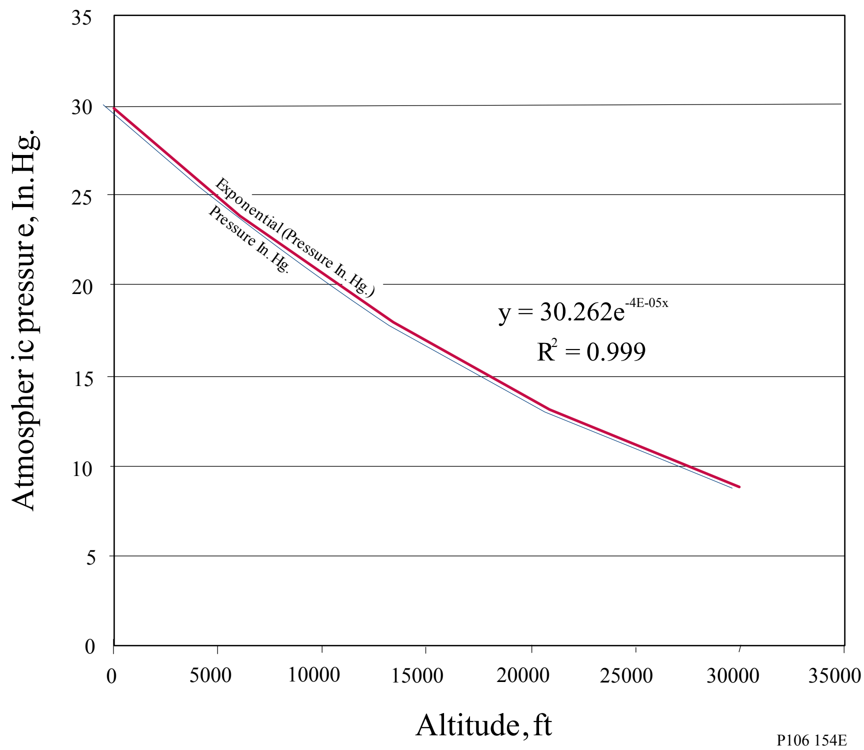
Atmospheric Properties:

Altitude		Atmospheric pressure, Psia @ 32 deg F	Absolute pressure Inches Mercury	Absolute pressure millimeters Mercury
Meters	Feet			
0	0	14.696	29.92	760
30	98	14.64	29.80	757
60	197	14.58	29.69	754
90	295	14.52	29.57	751
120	394	14.47	28.45	748
150	492	14.41	29.34	745
180	591	14.35	29.22	742
210	689	14.30	29.11	739
240	787	14.24	28.99	736
270	886	14.18	28.88	734
300	984	14.13	28.76	731
330	1083	14.07	28.65	728
360	1181	14.02	28.54	725
390	1280	13.96	28.43	722
420	1378	13.91	28.32	719
450	1476	13.85	28.20	716
480	1575	13.80	28.09	714
510	1673	13.74	27.98	711
540	1772	13.69	27.87	708
570	1870	13.64	27.76	705
600	1969	13.58	27.65	702
900	2953	13.06	26.59	675
1200	3937	12.55	25.56	649
1500	4921	12.07	24.57	624
1800	5906	11.60	23.63	600
2100	6890	11.16	22.71	577
2400	7874	10.73	21.84	555
2700	8858	10.31	20.99	533
3000	9843	9.91	20.18	513
3300	10827	9.53	19.40	493
3600	11811	9.16	18.65	474
3900	12795	8.81	17.93	456
4200	13780	8.47	17.24	438
4500	14764	8.14	16.58	421
6000	19685	6.69	13.61	346

Appendix F3-Influence of generic altitude on atmospheric pressure

Influence of Generic Altitude on Atmospheric Pressure

Atmospheric pressure vs altitude



Data showing atmospheric pressure vs altitude

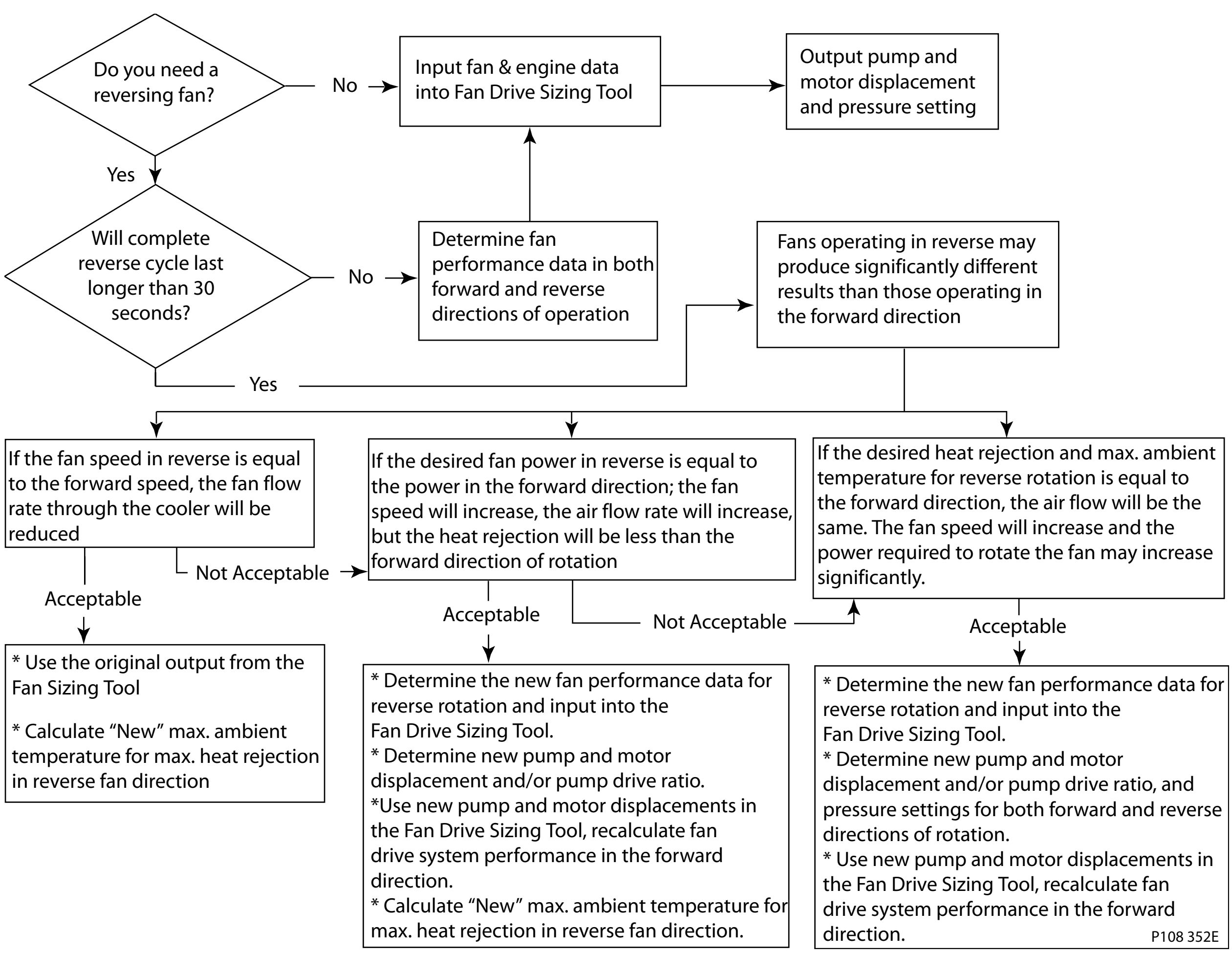
Data from ASHRAE Guide:		**Calculated pressure in. Hg**	**Error = calculated - given pressure in. Hg**
Altitude, ft.	**Pressure in. Hg**		
-1000	31.02	31.50	0.48
-500	30.47	30.87	0.40
0	29.921	30.26	0.34
500	29.38	29.66	0.28
1000	28.86	29.08	0.22
5000	24.89	24.78	-0.11
10000	20.58	20.29	-0.29
15000	16.88	16.61	-0.27
20000	13.75	13.60	-0.15
25000	11.1	11.13	0.03
30000	8.88	9.11	0.23
35000	7.04	7.46	0.42
40000	5.54	6.11	0.57
45000	4.36	5.00	0.64
50000	3.436	4.10	0.66

Appendix G-Influence of reversed fan rotation on system performance

In applications where it is necessary to change the direction of fan rotation, most fans will have reduced performance. Some fans are designed for equal, or similar, performance in the reverse direction, but this is not typical of many axial flow fans that are installed in vehicles today. Some fan blade designs have significantly different efficiencies depending on their direction of rotation and pitch angle (angle of attack). In this instance, we're using the term "efficiency" to indicate that the fan does not produce the same amount of air flow at the same speed; and/or it requires more power to produce the same amount of air flow as in the forward direction of rotation.

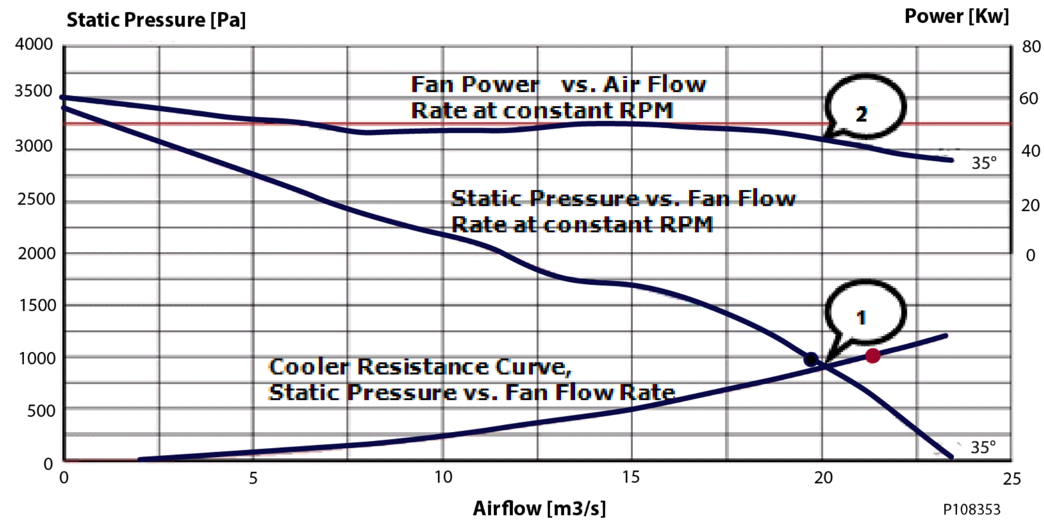
The cooling system designer is encouraged to consult with their fan supplier for performance characteristics specific to their fan; based on the speed of operation, static pressure drop, and direction of rotation. Using this information, it is possible to validate whether the fan will meet the specific needs of the cooling system. Given this information, the system designer can investigate the effect of these changes and determine if corrective actions need to be introduced.

The following is a guide to some of the characteristics that one could encounter when reversing the direction of a fan. For these purposes, the fan is assumed to be operating at its design "trim" speed and at the desired maximum "static" operating pressure across the cooler when in the forward direction.



Appendix G-Influence of reversed fan rotation on system performance

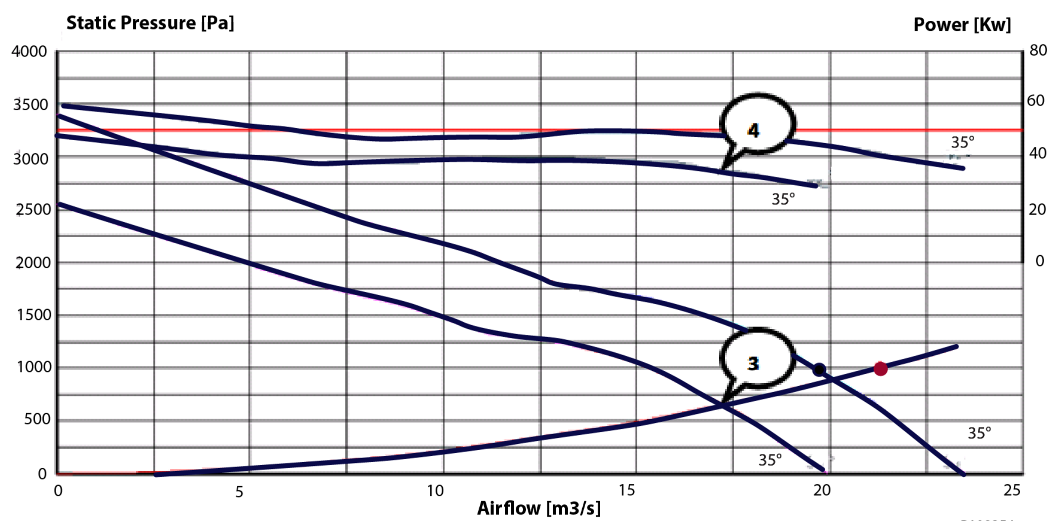
Baseline: Fan operating in Forward Direction at 2600 RPM



P108353

1. The Flow Rate for a given fan speed is determined by the intersection of the Fan's Flow/Static-Pressure Curve for that speed and the Cooler's Resistance Pressure vs Airflow characteristic curve.
2. The power required to turn the fan at that speed (airflow & static pressure) is located directly above this intersection point on the Fan Flow Rate vs Fan Power Curve at constant RPM. The magnitude of the fan power is determined by the secondary Y-Axis in the upper right hand corner of the graph.

Condition 1: Fan operating in Reverse Direction at 2600 RPM

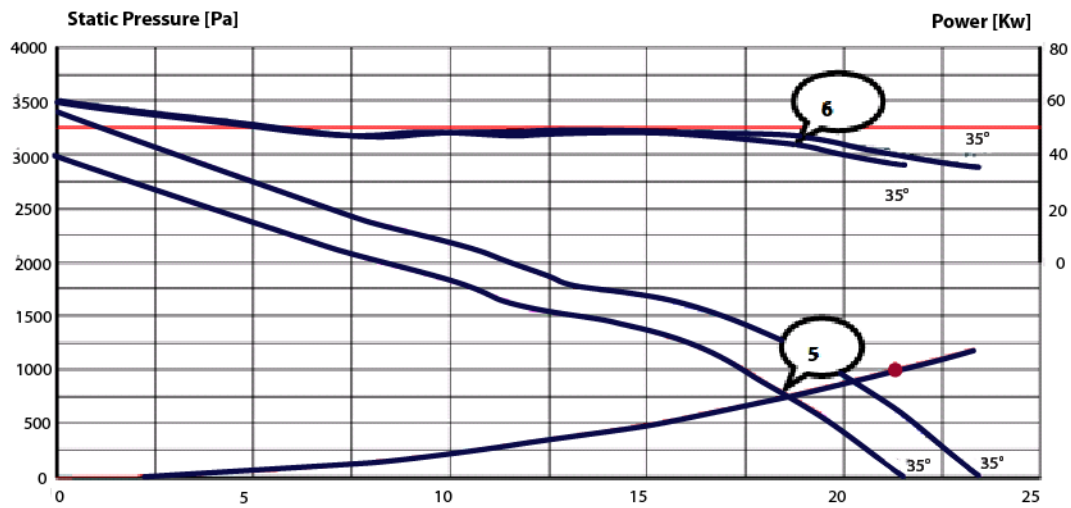


P108354

1. The Flow Rate for 2600 RPM in the reverse direction is less than that produced by the fan at 2600 RPM in the forward direction of rotation. Since the air flow rate is less, the static pressure across the cooler is also reduced.
2. The power required to turn the fan at the same speed, but in reverse direction, is less than that required in the forward direction of rotation. The fan is producing less flow at a lower static pressure; thus a lower power requirement.

Appendix G-Influence of reversed fan rotation on system performance

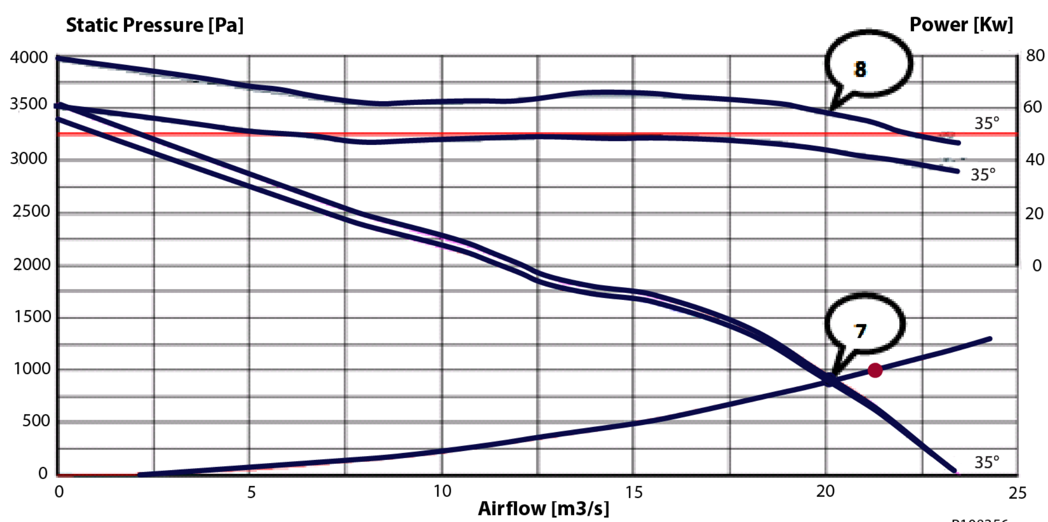
Condition 2: Fan operating in Reverse Direction at approximately the same input power level (by increasing the fan speed in reverse)



P108355

1. The fan is producing less flow, even though the fan speed is now 2810 RPM in the reverse direction of rotation.
2. The power required to turn the fan at 2810 RPM, in reverse direction, is approximately the same as that required to turn the fan at 2600 RPM in the forward direction of rotation. The fan is still producing less flow at a lower static pressure. (Fan efficiency is reduced in the reverse direction.)

Condition 3: Fan operating in Reverse Direction at approximately the same flow rate (by increasing the fan speed in reverse)

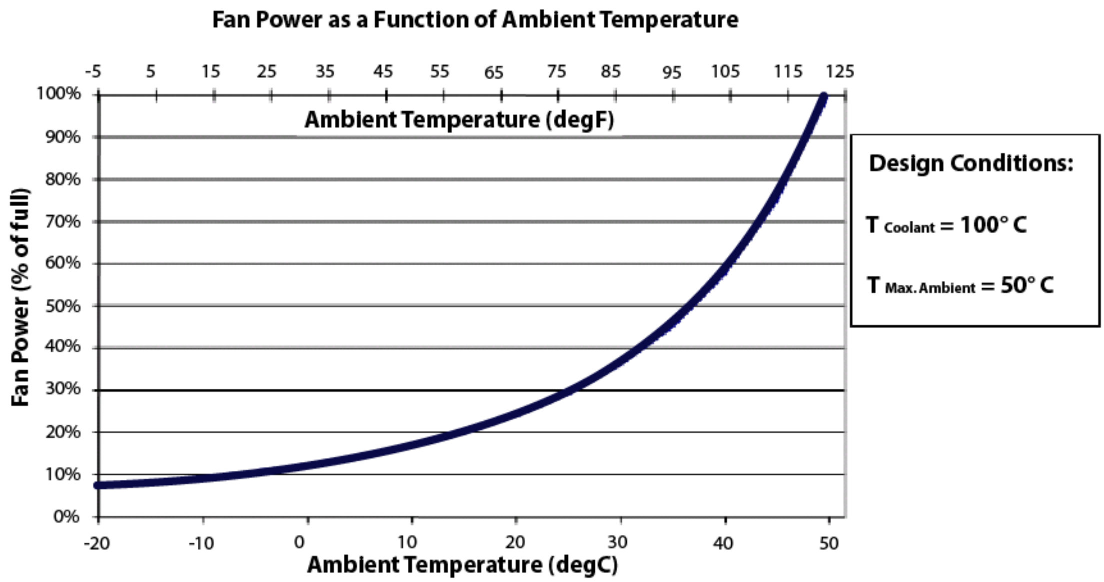


P108356

1. The fan is producing approximately the same flow rate, even though the fan speed is now 3060 RPM in the reverse direction of rotation.
2. The fan requires more power at the same flow rate and static pressure, because the fan is less efficient in the reverse direction of rotation.

Appendix G-Influence of reversed fan rotation on system performance

System Response to Reversed Fan Rotation:



$$\frac{T_{Coolant} - T_{ActualAmb}}{T_{Coolant} - T_{Max.\ Ambient}} = \text{Cooling\_Effectiveness}$$

P108357

- "Cooling Effectiveness" is proportional to the cooling capacity of the cooler; i.e., the ratio is greater than one when the $_{\text{Actual Amb}}$ient Temperature is less than the original design Maximum Ambient Temperature (System Design Temperature). When the ratio is greater than 1.0, the cooler will not require as much air flow to satisfy the cooling needs of the system. Therefore, the Fan Power requirement will be less than at the "design" maximum ambient temperature.
- At the "design" Maximum Ambient Temperature, reducing the air flow rate thru the cooler has an equivalent effect of reducing the "Cooling Effectiveness" of the cooler; as such, the two concepts may be equated.
- Therefore, as the air flow rate thru the cooler is reduced (Cooling Effectiveness ≤ 1.0); a new "$T_{\text{Actual Amb}}$" can be calculated, which will define a new upper limit of ambient temperature where the cooler will still satisfy the maximum heat load.

The new "$T_{\text{Actual Amb}}$" temperature will be lower than the original Design Maximum Ambient Temperature.

Appendix G-Influence of reversed fan rotation on system performance

Derivation:

$$\text{Heat Rejection}_{Actual} = \dot{m}_{\text{Actual}} * C_P * \left(T_{Coolant} - T_{\text{Actual Amb}}\right)$$

$$and$$

$$\text{Heat Rejection}_{\text{Maximum}} = \dot{m}_{\text{Maximum}} * C_P * \left(T_{Coolant} - T_{\text{Maximum Actual Ambient}}\right)$$

$$\left(\frac{\text{Heat Rejection}_{Actual}}{\text{Heat Rejection}_{\text{Maximum}}}\right) = \left(\frac{Q_{\text{Actual}} * \left(T_{Coolant} - T_{\text{Actual Amb}}\right)}{Q_{Maximum} * \left(T_{Coolant} - T_{\text{Maximum Actual Ambient}}\right)}\right)$$

$$IF$$

$$\left(\frac{\text{Heat Rejection}_{Actual}}{\text{Heat Rejection}_{\text{Maximum}}}\right) \equiv 1.0, \text{ and } Q_{\text{Actual}} \leq Q_{Maximum}$$

$$then$$

$$\left(\frac{Q_{\text{Actual}}}{Q_{Maximum}}\right) \leq 1.0, and$$

$$\left(\frac{T_{Coolant} - T_{ActualAmb}}{T_{Coolant} - T_{MaxAmbient}}\right) \text{must be} \geq 1.0, \text{ therefore the new } T_{\text{ActualAmb in Rev}} \text{ must be} \leq T_{MaxAmbient}$$

$$Let:$$

$$\left(\frac{Q_{\text{Actual}}}{Q_{Maximum}}\right) = 0.8424 = \left(\frac{\text{Flow at 2600 RPM Reverse}}{\text{Flow at 2600 RPM Forward}}\right)$$

$$\text{And Let}:$$

$$\left(\frac{\text{Heat Rejection}_{Actual}}{\text{Heat Rejection}_{\text{Maximum}}}\right) \equiv 1.0,$$

$$T_{Coolant} = 100^{\circ} C$$

$$T_{MaxAmbient} = 50^{\circ} C$$

$$Therefore,$$

$$\left(T_{Coolant} - T_{\text{Maximum Actual Ambient}}\right) * \left(\frac{Q_{Maximum}}{Q_{\text{Actual}}}\right) = \left(T_{Coolant} - T_{\text{New Actual Amb in Rev}}\right)$$

$$T_{\text{New Actual Amb in Rev}} = \left(\frac{1}{0.8424}\right) * 100^{\circ} C - \left(\frac{1}{0.8424} - 1\right)\left(100^{\circ} C - 50^{\circ} C\right) = 40.65^{\circ} C$$

Suggested Guidelines for Reversing Fan Drive Systems:

IF the desire is to keep the *fan speed* in the reverse direction the same as in the forward direction:

This will occur automatically, IF the fan drive system is flow limited by pump speed, or displacement; but not IF the system is pressure limited.

Appendix G-Influence of reversed fan rotation on system performance

- The heat rejected by the cooling system will be less than in the forward direction of rotation,
- The power to drive the fan in reverse will be less than in the forward direction,
- The Torque to turn the fan, and therefore the delta pressure across the hydraulic motor, will be proportional to the power ratio, since the fan speed is the same.

IF the desire is to keep the *fan power* in the reverse direction the same as in the forward direction:

- The Heat Rejection by the system will still be less than in the forward direction, but better than when maintaining constant fan speed, (HR α Speed Ratio)
- The fan speed in reverse will be greater than in the forward direction. The Torque to turn the fan (and hydraulic delta pressure) will be less than in the forward direction, (Power to turn the fan is the same, but the speed is greater.)
- This condition may define the maximum pump displacement or speed required to satisfy the flow requirements of the fan system, since the flow required to rotate the fan will be larger in the reverse direction than in the forward direction.
- Be sure to validate these conditions with the fan supplier. The new speed requirement may exceed the maximum speed limit.

Tip speed is proportional to the fan rotational speed and internal stresses in the fan are proportional to the square of the fan speed.

IF the desire is to keep the *fan flow rate* in the reverse direction the same as in the forward direction:

- The Heat Rejection by the system will be the same as in the forward direction,
- The Power required to rotate the fan in the reverse direction will be greater than in the forward direction of rotation, because the fan is less efficient in the reverse direction of rotation,
- The fan speed in reverse will be greater than in the forward direction. The Torque to turn the fan (and hydraulic delta pressure) may be greater than in the forward direction, (Power to turn the fan is greater, and the speed is greater.)
- This condition may define the maximum pump displacement required to satisfy the flow requirements of the fan system, since the flow required to rotate the fan will be larger in the reverse direction than in the forward direction.
- Be sure to validate these conditions with the fan supplier. The new speed requirement may exceed the maximum speed limit.

Tip speed is proportional to the fan rotational speed and internal stresses in the fan are proportional to the square of the fan speed.

Appendix G-Influence of reversed fan rotation on system performance

Performance Summary: (For Reference Only)

Refer to performance curves in previous graphs

Direction of Rotation	Forward	Reverse		
Fan Speed, RPM	2600	2600	2810	3060
	1	3	5	7
Air Flow Rate, m^3/s	20.1	17.2	18.6	20.3
Static Pressure, Pa	894	653	763	904
Total Pressure, Pa	1450	1060	1240	1460
	2	4	6	8
Fan Power, kW	43.7	34.8	43.9	56.8
Total Efficiency, %	67%	52%	52%	52%

P108359

Appendix H-System considerations for H1 fan drives with reversed fan rotation

System Considerations for H1 Fan Drives with Reversed Fan Rotation

As stated in Appendix G, in applications where it is necessary to change the direction of fan rotation, most fans will have reduced performance. For fan systems using H1 pumps with the H1 Fan Drive Control Valve, the control valve receives a command from the PLUS+1® Controller to regulate fan speed by changing pump displacement. When the pump speed is at the design set point, fan speed required for the desired cooling capacity is determined by the pump's displacement.

In many systems, the fan's speed at this condition may be close to its maximum design speed. If the pump speed increases beyond the set point and the pump is commanded to maximum displacement; then the fan speed will increase in proportion to the pump speed ratio and the delta pressure across the fan motor will increase in proportion to the square of the pump speed ratio.

To prevent this from happening, the system designer is encouraged to limit the maximum fan speed in each direction of rotation by adjusting the set pressure of the Pressure Limiters for both directions of rotation.

Pressure Limiter Adjusting Procedure

For the forward direction of rotation, determine the delta pressure across the fan motor when it's at the maximum desired fan speed and power; this is the *forward design pressure*. Adjust the pressure limiter setting to be equal to the *forward design pressure plus 20 bar* (to account for piece-to-piece variation within the cooling system) for the forward direction of rotation.

For the reverse direction of rotation, determine the delta pressure across the fan motor when it's at the maximum desired fan speed and power; this is the *reverse design pressure*, (consult with the fan and cooler manufacturer to determine the desired fan speed and power for the fan when it is rotating in the reverse direction, reference Appendix G). Adjust the pressure limiter setting to be equal to the *reverse design pressure plus 20 bar* (to account for piece-to-piece variation within the cooling system) for the reverse direction of rotation.

When the pump arrives at the OEM's production line, it will have a nominal PL setting of 150 bar and may have a High Pressure Relief Valve setting of 250 bar, 300 bar, 350 bar, or 400 bar. depending on order code selected. The PL must be re-adjusted to ensure that the fan reaches the desired fan speed to satisfy the cooling needs of the system. It may be necessary to replace the HPRV with a valve that has a higher pressure setting. To replace the valves, follow the recommended procedures that are available in Service Documents 520L0848 or 520L0958.

The HPRV setting must be greater than the PL setting by at least 30 bar after the PL has been re-adjusted.

When adjusting the pressure limiters, follow the recommended adjustment procedure that is available in Service Document 520L0848 or 520L0958, or after stopping the prime mover and ensuring that the fan has stopped rotating.

Verify that the pressure limiter settings will not cause the fan speed to exceed the manufacturer's maximum speed limitation when the engine's speed is at its No Load High Idle (NLHI) condition and during periods of positive acceleration between the design set point and NLHI. Make adjustments accordingly. Refer to the graphs on pages 58-59 for typical representative behavior of fan speed changes when the engine is accelerating.

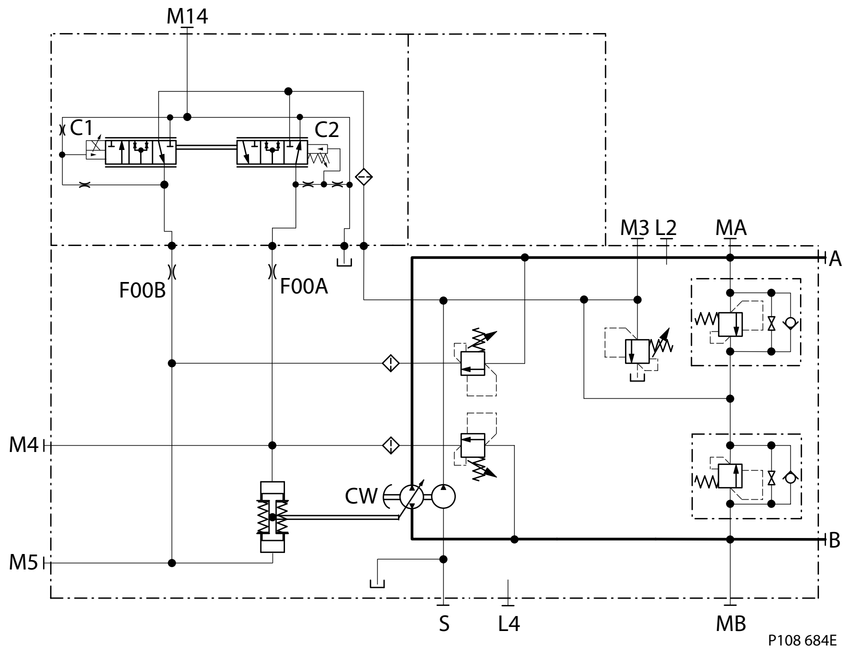
In the event of sudden engine speed increases, when the fan is operating close to the Pressure Limiter and High Pressure Relief Valve settings, then it is possible for the fan speed to exceed the design trim speed by approximately 10% (System delta pressure will exceed the setting of the High Pressure RV by approximately 20%.) Consequently, the system integrator needs to take this into consideration when sizing the fan and hydraulic components.

Pump and motor life are influenced by both speed and pressure. Contact your Danfoss technical representative for an analysis of the systems' duty cycle to determine if it will meet the desired life goals.

Appendix H-System considerations for H1 fan drives with reversed fan rotation

Additional Information concerning the H1 Fan Drive Controller Option

H1 Pump with FDC Control - Schematic



Flow Direction vs. Signal

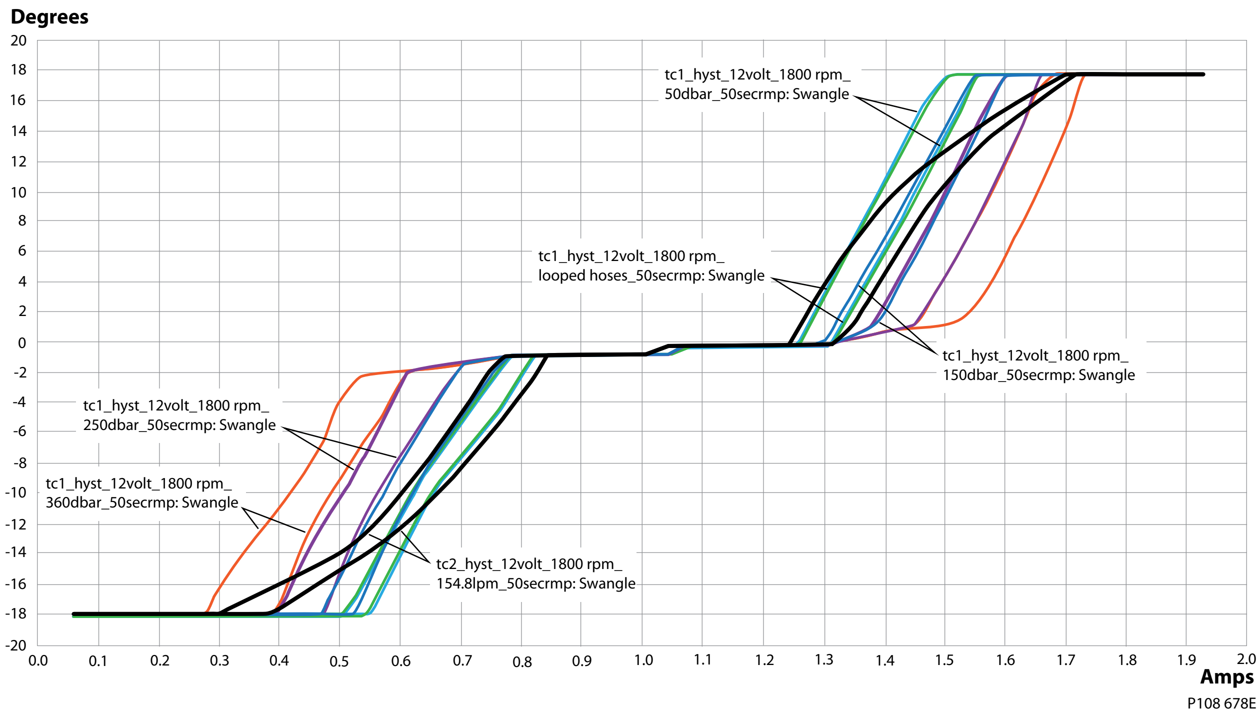
	Pump Rotation Clockwise (CW) as seen from shaft		Pump Rotation Counterclockwise (CCW) as seen from shaft	
	Forward fan rotation	Reverse fan rotation (radiator cleaning)	Forward fan rotation	Reverse fan rotation (radiator cleaning)
Control current	Less than 1050 mA for 12 Vdc Less than 550 mA for 24 Vdc	More than 1050 mA for 12 Vdc More than 550 mA for 24 Vdc	Less than 1050 mA for 12 Vdc Less than 550 mA for 24 Vdc	More than 1050 mA for 12 Vdc More than 550 mA for 24 Vdc
System port A flow	In (Low)	Out (High)	Out (High)	In (Low)
System port B flow	Out (High)	In (Low)	In (Low)	Out (High)
Servo gauge port high pressure	M5	M4	M5	M4

FDC Start and End Current

Typical FDC control curve in Fan Drive System, overlaid on FDC *constant pressure* control curves from a representative NFP *Load Valve* test. In the FDC Control test, a fixed orifice is placed in the flow stream between the A and B ports of the pump and it is sized to be equivalent to the flow/pressure relationship of a fixed displacement motor driving a fan.

Appendix H-System considerations for H1 fan drives with reversed fan rotation

Start and End Current



Typical, representative points on the curve are shown in the following table.

Start and End Current

Control	Parameter	Forward Current (mA) (decreasing current direction)	Reverse Current (mA) (increasing current direction)
12Vdc	Start Current (typical)	780	1300
	End Current	Dependent on Fan Drive System Pressure	
	Tolerance of End Current		
	Nominal Range		
	Max Current Allowed		1800
24Vdc	Start Current (typical)	400	680
	End Current	Dependent on Fan Drive System Pressure	
	Tolerance of End Current		
	Nominal Range		
	Max Current Allowed		920

Operating Envelope for H1 Pumps with Fan Drive Control

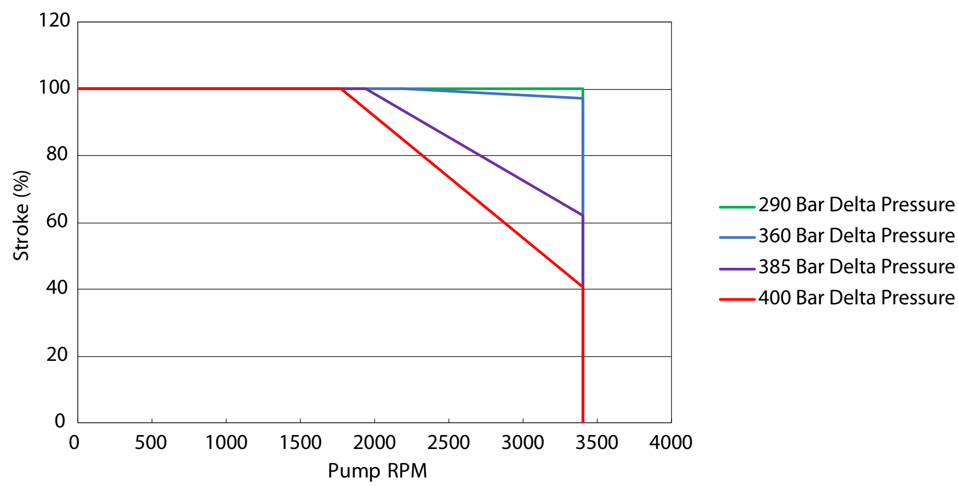
The Fan Drive Control (FDC) has limitations on the maximum servo delta pressure developed, compared to other types of controls, and so there are limitations to the operating conditions which can be achieved in the various frame size H1 pumps. Typical fan drive systems are unusual in that they achieve peak pressure only at high flows, so it is important that FDC equipped pumps not be applied beyond the limits defined below.

Typical limits for the operating envelope of the H1 pump family with the Fan Drive Control are shown in the following figures. There is one figure for each H1 pump displacement. Note that the constant pressure values are not the same for each displacement, as all are not necessarily applicable for each displacement.

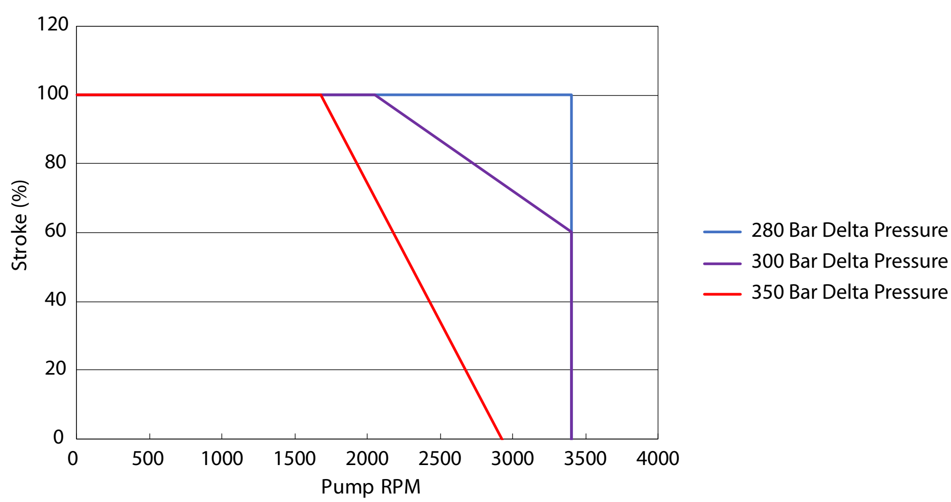
Appendix H-System considerations for H1 fan drives with reversed fan rotation

In the following figures, the area underneath each curve represents the appropriate operating envelope for a given system delta pressure:

H1P045 pump with CP8 valve plate and FDC

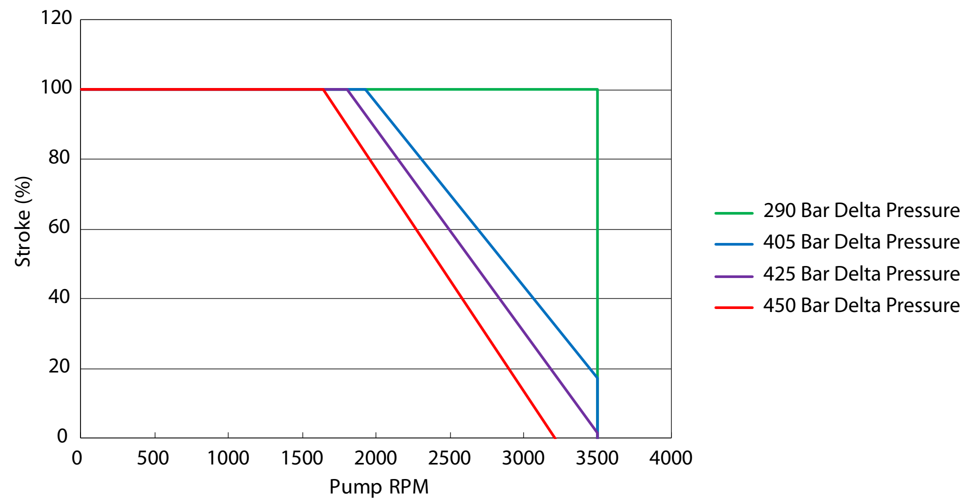


H1P053 pump with CP8 valve plate and FDC

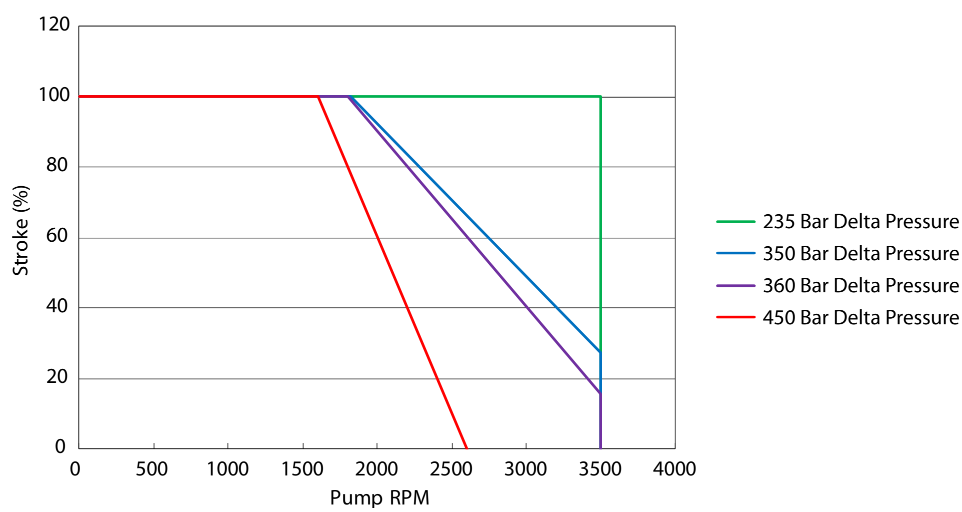


Appendix H-System considerations for H1 fan drives with reversed fan rotation

H1P060 pump with CP17 valve plate and FDC

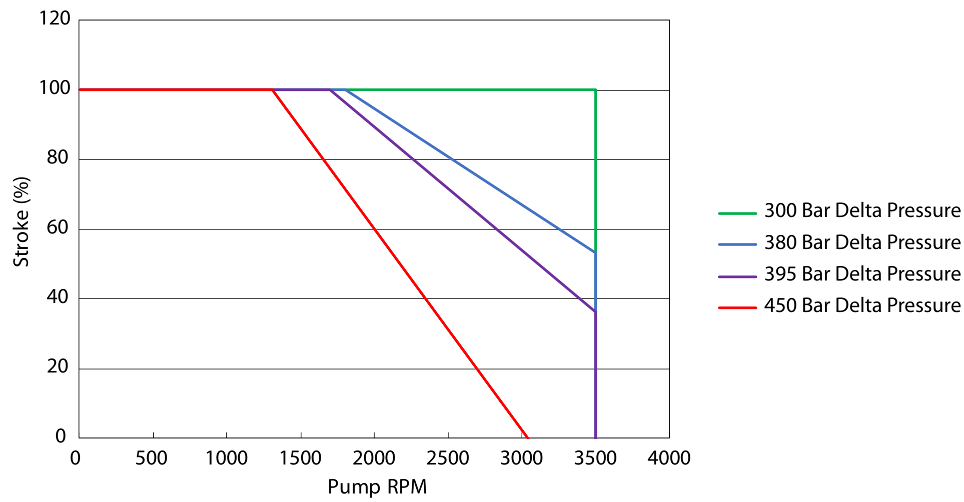


H1P068 pump with CP17 valve plate and FDC

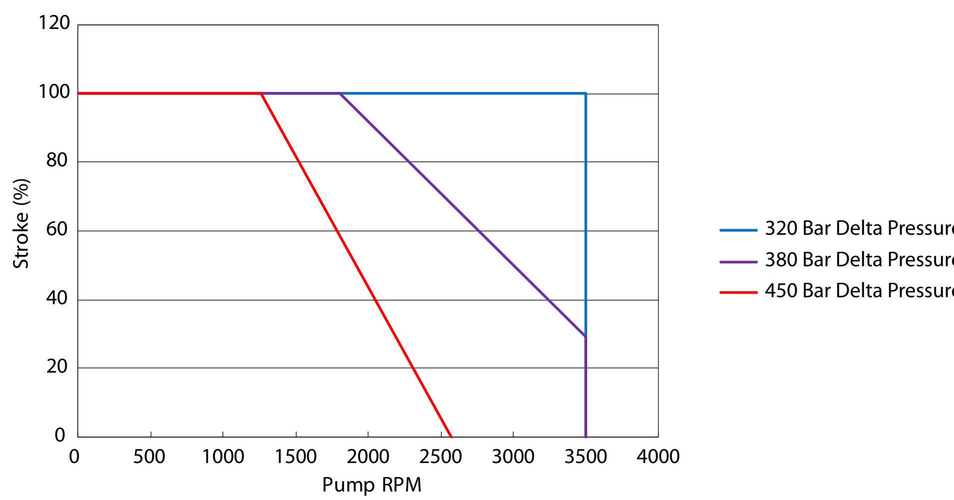


Appendix H-System considerations for H1 fan drives with reversed fan rotation

H1P069 pump with CP17 valve plate and FDC

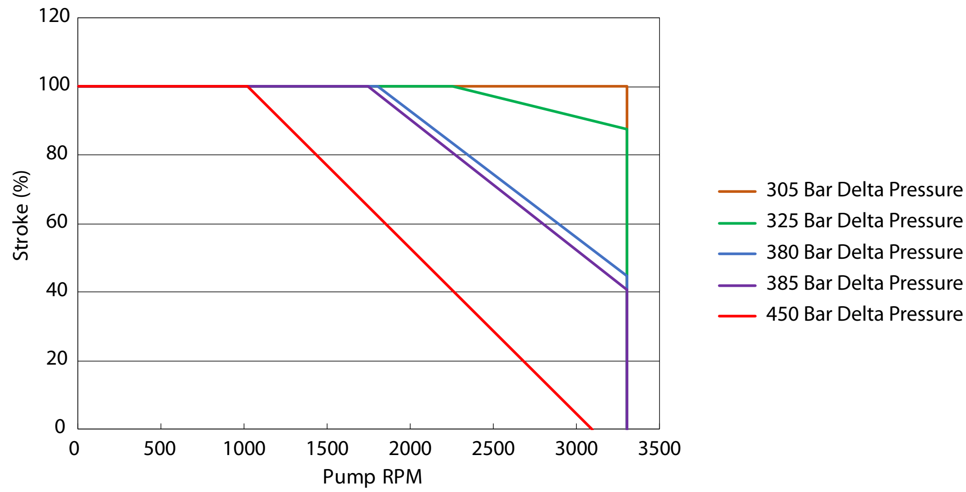


H1P078 pump with CP17 valve plate and FDC

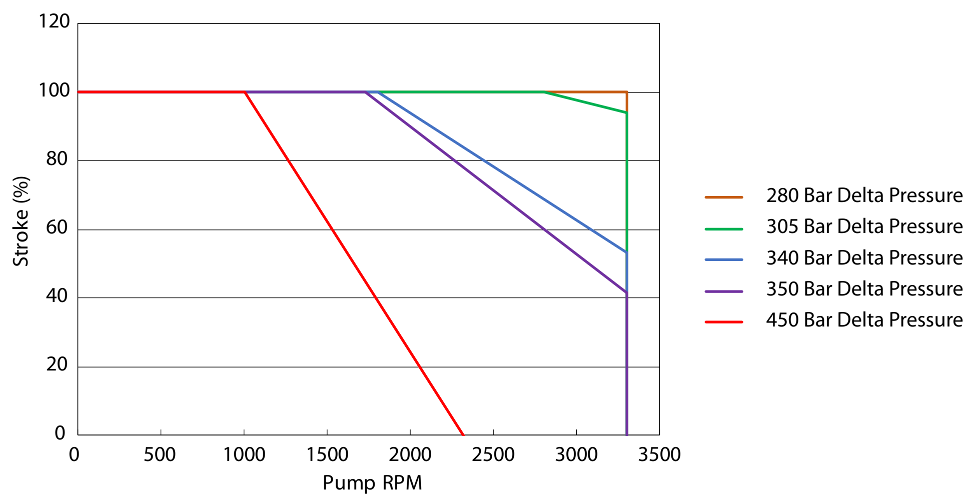


Appendix H-System considerations for H1 fan drives with reversed fan rotation

H1P089 pump with CP17 valve plate and FDC

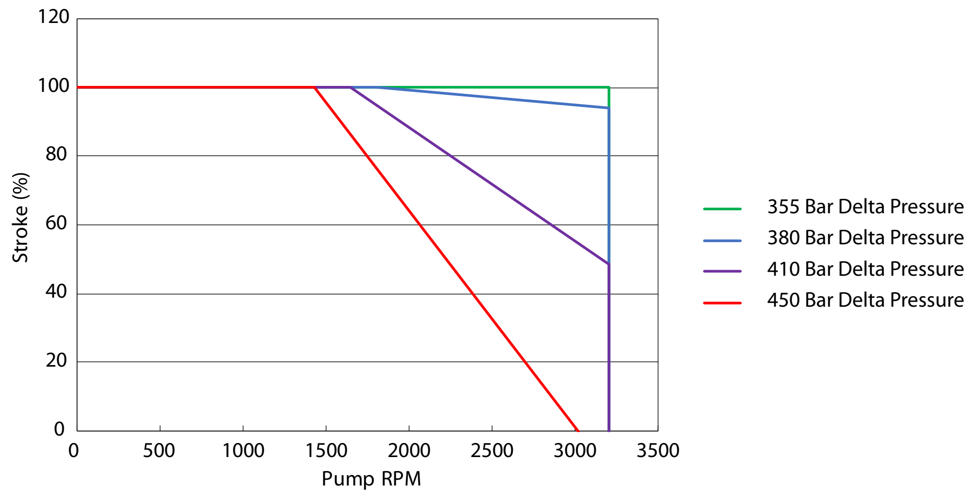


H1P100 pump with CP17 valve plate and FDC

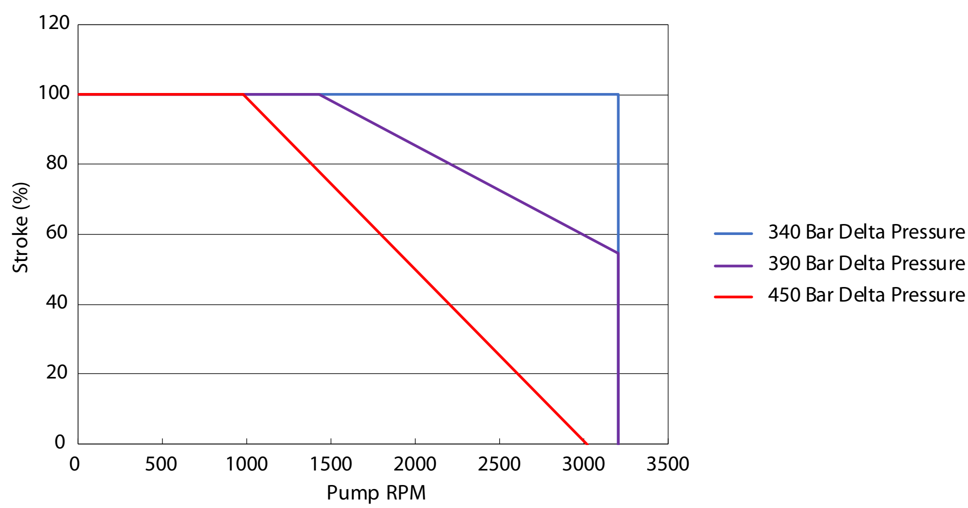


Appendix H-System considerations for H1 fan drives with reversed fan rotation

H1P115 pump with CP17 valve plate and FDC

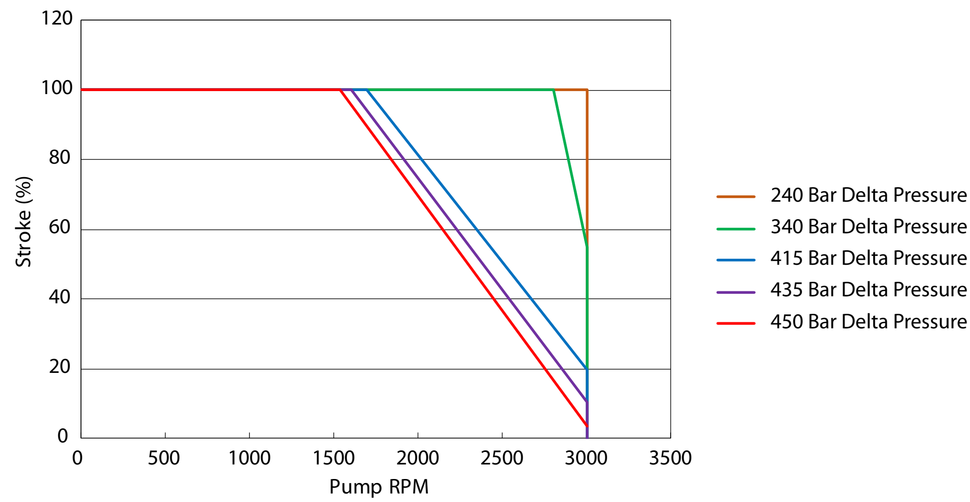


H1P130 pump with CP17 valve plate and FDC

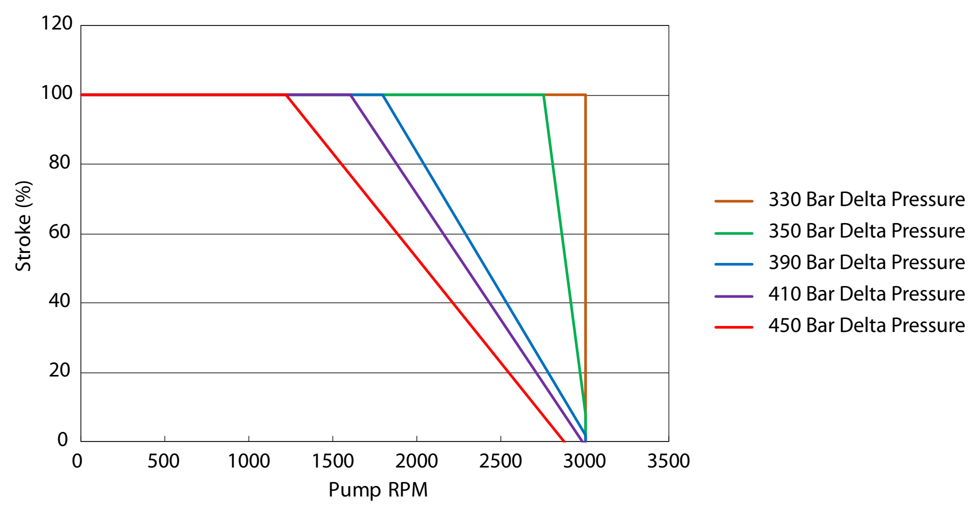


Appendix H-System considerations for H1 fan drives with reversed fan rotation

H1P147 pump with CP17 valve plate and FDC

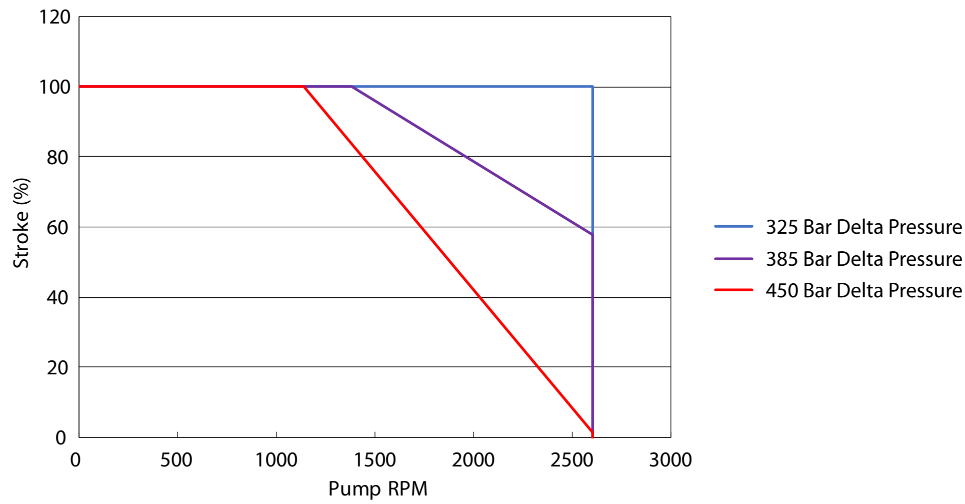


H1P165 pump with CP17 valve plate and FDC

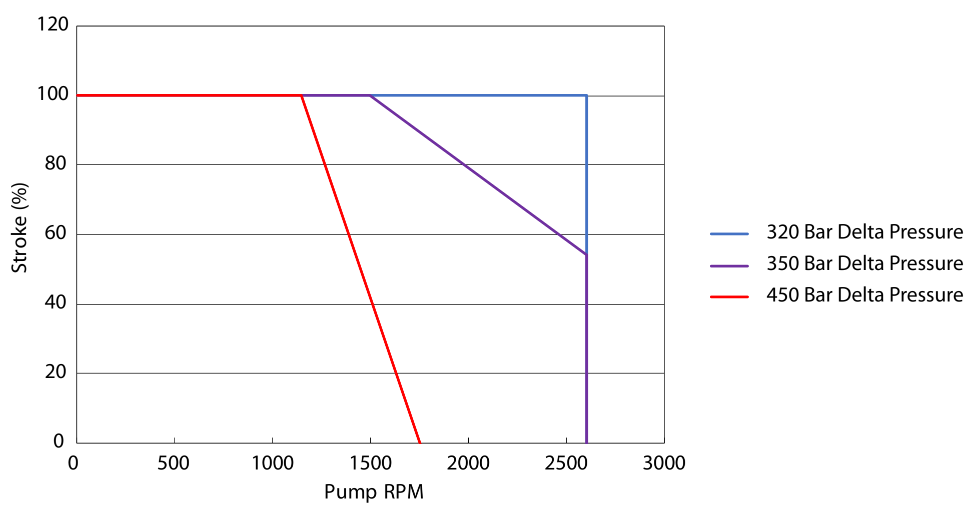


Appendix H-System considerations for H1 fan drives with reversed fan rotation

H1P210 pump with CP17 valve plate and FDC



H1P250 pump with CP17 valve plate and FDC



Sensitivity to Prime Mover Speed Changes (Load Sensitivity) - (J Frame as example)

The natural NFPE behavioral characteristic of the pump tends to correct for engine/pump speed variations which occur as a normal part of operation, and maintain the fan speed near to the level which was produced before the engine speed change. Of course, this can only occur to the extent that the pump can achieve its maximum displacement; beyond that, significant fan speed change will occur.

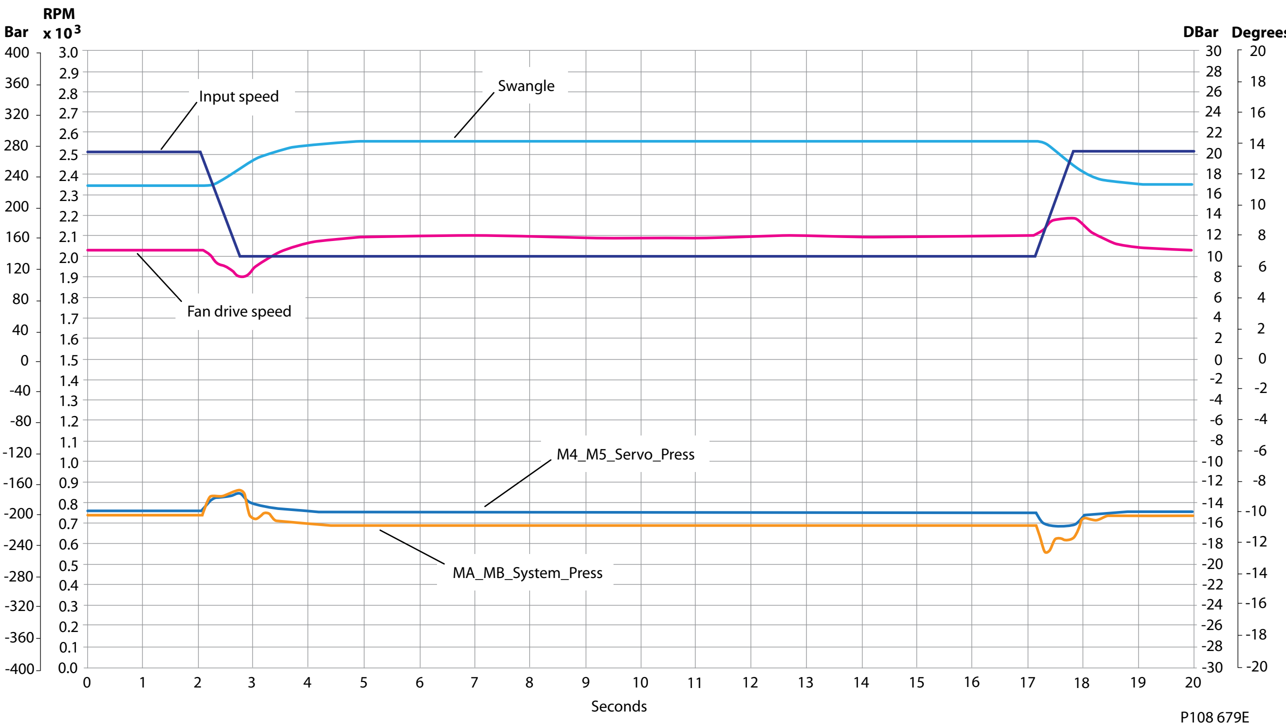
In the conditions representing an *engine lugging* condition (engine speed reduction less than 20%), there was no significant reduction in fan speed; the lower pump moments at the lower pump speed allowed the pump to increase stroke to maintain a relatively constant output flow.

In all conditions representing a *throttle down* condition (engine speed reduction greater than 50%), the percent of fan speed drop was lower than the percent of pump speed drop.

Curves shown below are from J frame testing, and are representative of the behavior which will be seen on other frame sizes.

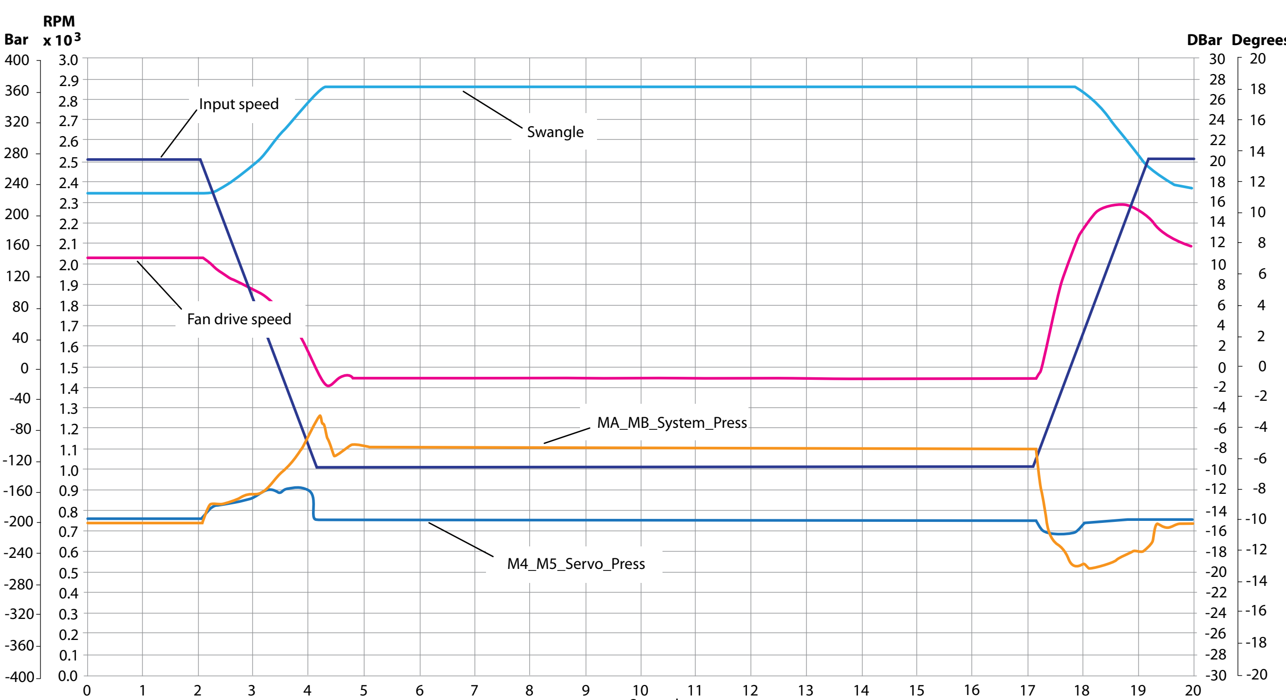
Appendix H-System considerations for H1 fan drives with reversed fan rotation

Prime Mover Speed Change (2500 - 2000 - 2500 rpm)



P108 679E

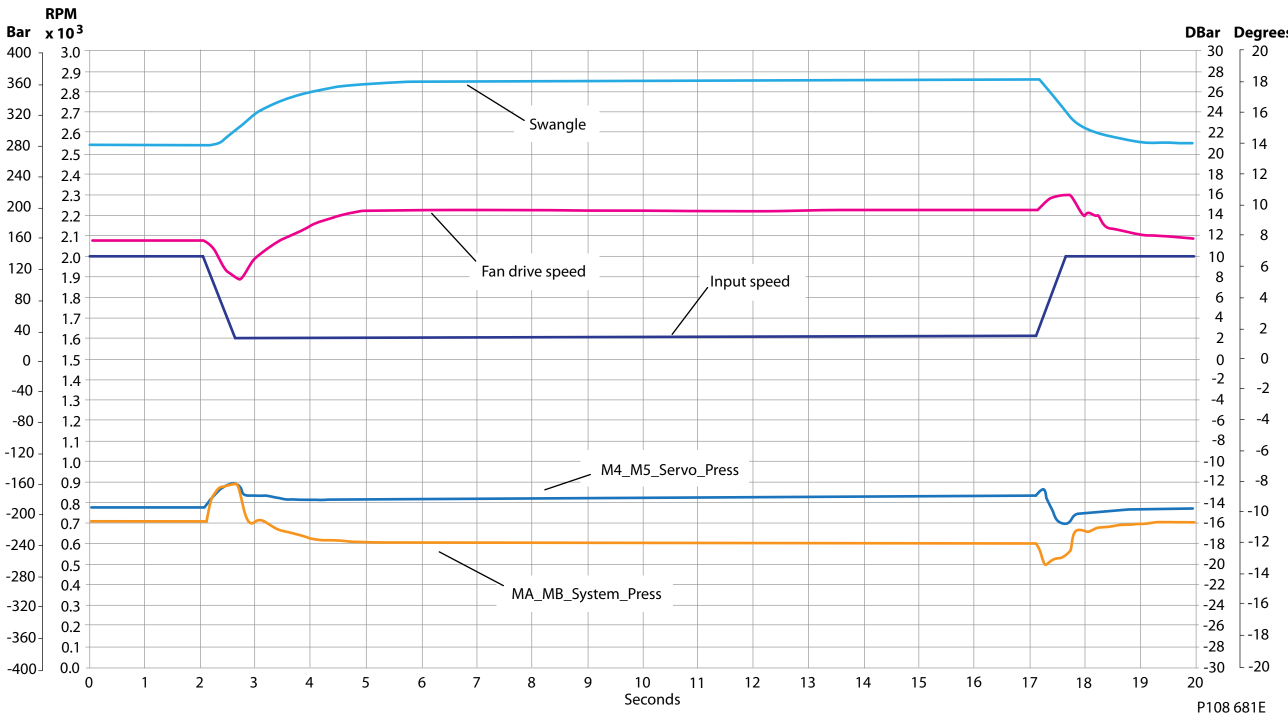
Prime Mover Speed Change (2500 - 1000 - 2500 rpm)



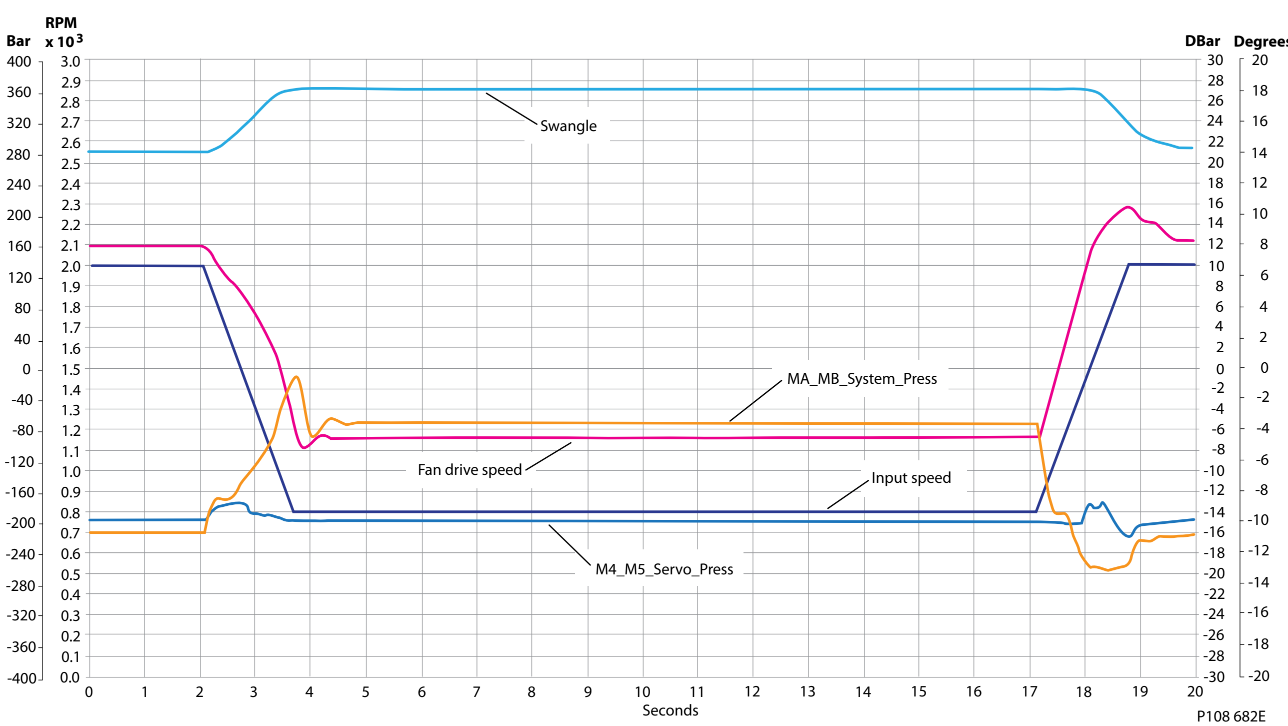
P108 680E

Appendix H-System considerations for H1 fan drives with reversed fan rotation

Prime Mover Speed Change (2500 - 1600 - 2500 rpm)



Prime Mover Speed Change (2500 - 800 - 2500 rpm)



Appendix H-System considerations for H1 fan drives with reversed fan rotation

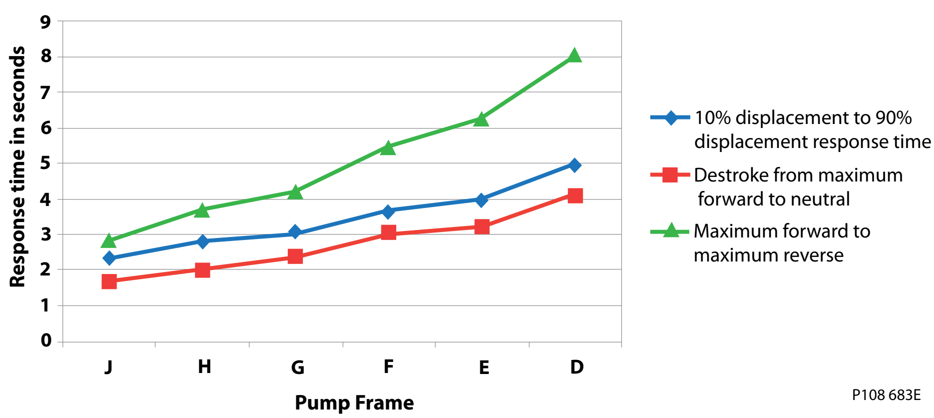
J Frame is used as an example only

In the event of sudden engine speed increases, when the fan is operating close to the Pressure Limiter and High Pressure Relief Valve settings, then it is possible for the fan speed to exceed the design trim speed by approximately 10% (System delta pressure will exceed the setting of the High Pressure RV by approximately 20%.) Consequently, the system integrator needs to take this into consideration when sizing the fan and hydraulic components.

Due to the failsafe functionality of the FDC control the pump will stroke to max. displacement in case the input signal to the pump control and the Diesel engine will be switched off at the same time. In this situation a low loop event can occur which may damage the pump. Therefore, it's strictly recommended to keep the input signal to the pump control alive while switching off the engine.

For further information please contact your Danfoss representative

H1 FDC Response Time (with Typical Fan Drive System Loading)



Warning

This control is for Fan Drive Systems only!

Use in other systems could result in unintended movement of the machine or its elements. Loss of the input signal to this control will cause the pump to produce maximum flow. Contact Danfoss or an authorized distributor with questions regarding the use of this product.

Application startup method (to account for PL offset)

The Pressure Limiter (PL) setting at high swashplate angles can be significantly (30 to 50 bar) lower than the setting on the Danfoss production test stand, which is done with system ports blocked.

This effect has a strong impact on Fan Drive applications; these applications are unique in that they always reach their peak pressure at a high swashplate angle (due to the typical fan speed/torque curve). Since the PL is needed to limit the peak system pressure, and therefore fan speed, in a particular application, this reduction in PL setting due to the standard factory test procedure will cause less than optimum cooling. Because of this behavior, the following procedure should be used to determine the nominal value for the factory PL adjustment (nomenclature) for a new fan drive application. This will be required if there is a desire to pre-set the Pressure Limiters at a test facility without a working fan assembly, and prior to installing the pump onto a machine:

1. Order the FDC pump with the standard PL settings of 150 bar. Standard pump configurations are provided with 150 bar PL settings and with the HPRV settings at 250 bar, 300 bar, 350 bar, and 400 bar.
2. Start the machine with a neutral signal to the FDC control and with the engine at its rated operating speed. Gradually change the signal to the FDC to increase the fan speed; if the fan begins to exceed its maximum operating speed limit, use the procedure mentioned in paragraph above or shut off the

Appendix H-System considerations for H1 fan drives with reversed fan rotation

machine and adjust the PLs to a lower pressure setting. Repeat this for forward and reverse directions of fan rotation; until it is assured that the PLs are limiting fan speed within acceptable levels.
This step is to ensure that the fan cannot operate above its maximum speed limit in subsequent steps. It may be necessary to increase the pressure setting of the HPRVs. Verify that the Pressure Limiter settings are at least 30 bar below the pump's High Pressure Relief Valves following the recommended installation procedure provided within Service Bulletin **520L0848** or **520L0958**.

3. Adjust the PL settings to the value determined during the sizing process by following the recommended adjustment procedure provided within Service Bulletin **520L0848** or **520L0958**, or stop the machine and ensure that the fan has stopped.

   This will be a mid-range current level, not zero amps.

4. Operate the engine at its rated operating speed, and with a maximum command to the FDC. If necessary, adjust the PLs to achieve the desired peak fan speed as above following the recommended adjustment procedure provided within Service Bulletin **520J0848** or **520L0958** (do this for forward and then for reverse directions of fan rotation).
5. During the Application Review process, or at "Initial Application Start" follow the instructions in steps 2 through 4. Stop the machine, then disconnect the pump system hoses from the fan drive circuit, and block the system ports to simulate the factory test stand -"no flow" condition.
6. Start the machine with a neutral signal to the FDC, and operate at the following conditions to simulate the factory test stand conditions: 1775 rpm pump speed, 50C - 80C oil temperature, and at zero and full flow current to the FDC solenoid. Record the System Delta Pressure achieved for the forward and reverse flow directions.
7. Verify that these system delta pressure settings are at least 30 bar below the pump's High Pressure Relief Valves (if not, stop the machine, consult Danfoss technical sales and then install the next highest pressure level HPRVs, and repeat this test) following the recommended installation procedure provided within Service Bulletin **520L0848** or **520L0958**

   The PL setting must be less than the HPRV setting.

Appendix I-System Considerations for RDM Fan Drives

System Considerations for RDM Fan Drives

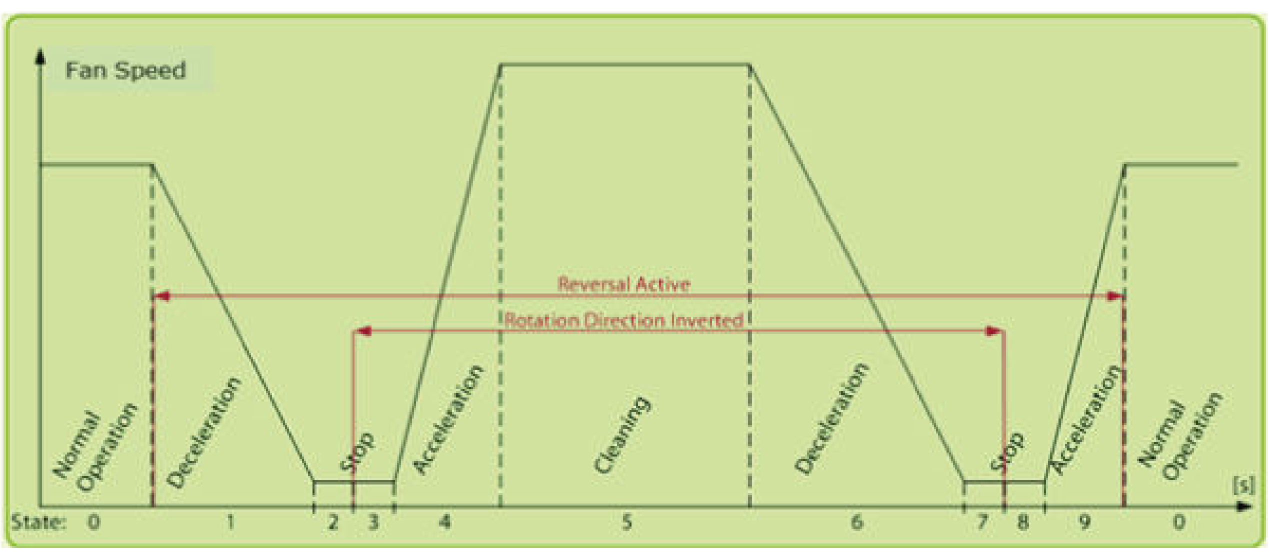
These application guidelines are for open circuit systems which are equipped with the Reverse Displacement Motor (RDM) to allow reversing of the load output direction without a directional control valve and the system losses that go with a DCV. The RDM should only be used in open circuit systems that regulate load speed by controlling system pressure based on the input command. A displacement, or flow controlled pump should not be used with the RDM; due to the risk of over speeding the motor and/or load as the direction changes. This is because the motor displacement goes from full forward through zero to full reverse.

Reversing Sequence

The recommended reversing sequence is shown below. The load is first decelerated to the minimum normal system speed by commanding the system to the lowest normal system pressure (low pressure standby) and allowing the load's speed to slow down. The RDM is then shifted to reverse and the system pressure is then increased to achieve the desired speed. The same process is then used to return the load to the original direction. The primary purpose of decelerating the load prior to shifting the motor is to reduce the "flywheel" affect inherent in reversing a spinning load. This will minimize pressure spikes and other disturbances in the system caused by feeding the energy of the rotating load back into the hydraulic system. It will also minimize the potential for requiring oil to flow through the motor system return line in a direction opposite that of the normal flow.

Another reason for reducing the speed and pressure in the system prior to initiating a motor reversal is to minimize the impact of the shifting process on the life of the motor. The higher the system pressure and/or speed is when the reversal is started, the greater the risk of wear in the swashplate bearings and a reduction in the life of those bearings.

Reversing Sequence



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If, for some reason, this ideal process can't be achieved, based on the system requirements for a faster load deceleration; the minimum requirement is that the system pressure setting be reduced prior to the reversal to minimize and delay the pressure spikes in the system while the fan is reversing. The delay in this step, between the pressure reduction signal and the reversing signal should be long enough to ensure the pressure control responds and the fan decelerates to the minimum speed condition. Experience to date would suggest that a 1 second delay is sufficient, but this should be verified by test. Once the fan reversal is accomplished, the pressure can be increased to achieve the desired fan speed, or fan acceleration time.

Since the unrestricted response time of the motor to shift from forward to reverse is approximately 350 ms, the delay required between sending the reversing signal and sending the signal to increase the

Appendix I-System Considerations for RDM Fan Drives

system pressure should be greater than 500 ms. A command to increase system pressure before the motor reaches full displacement will result in higher system pressures while the fan is accelerating. A longer delay will allow the fan to decelerate more and will minimize the pressure to fully stop the fan and reverse its direction; but will also lead to reduced cooling capacity during the shift sequence. Likewise, when the motor is returning to the forward direction, the minimum delay time between these two events is 1 second, instead of the 500 ms interval above. This is because the natural shift response of the motor is slower when returning to the forward direction.

A duty cycle that follows these recommended guidelines will result in a reversing cycle “capability” of more than 60,000 cycles, because the motor is not actually shifting under high pressure. A duty cycle where the motor shifts before slowing down will see higher system pressures because the load energy is being dissipated more quickly. This will have an impact on the life of the motor kit, swashplate bearings and shaft bearings.

Shift Rate Control

In general, the best option is to allow the motor to shift as quickly as possible when the motor is at minimum speed. This will reduce the risk of the motor shifting under load, thus impacting the life of the swashplate bearings. There may be systems, however, where it is desired to control the shift rate of the motor. One example might be in systems with very high inertia loads.

If required, the reversal rate of the RDM can be controlled. The motor is equipped with an integrated shift valve, by controlling the input signal to the motor; the rate of reversal can be controlled. The shifting of the motor from forward to reverse happens through the range of 300-750 mA input for a 12 volt valve and 150-375 mA for a 24 volt valve. The input current should be ramped through this range based on the desired shift time of the motor. The return to the forward direction will happen through the range of 550-150 mA for a 12 volt valve and 275-75 mA for a 24 volt valve.

For motors with an integral shift valve, if the electrical input to the motor control can’t be regulated as described above; there is an option to install an orifice, either in front of the shift valve, or between the shift valve and the servo cavity. The first orifice will impact the forward-to-reverse shift without influencing the return-to-forward shift; the second will impact both reversals.

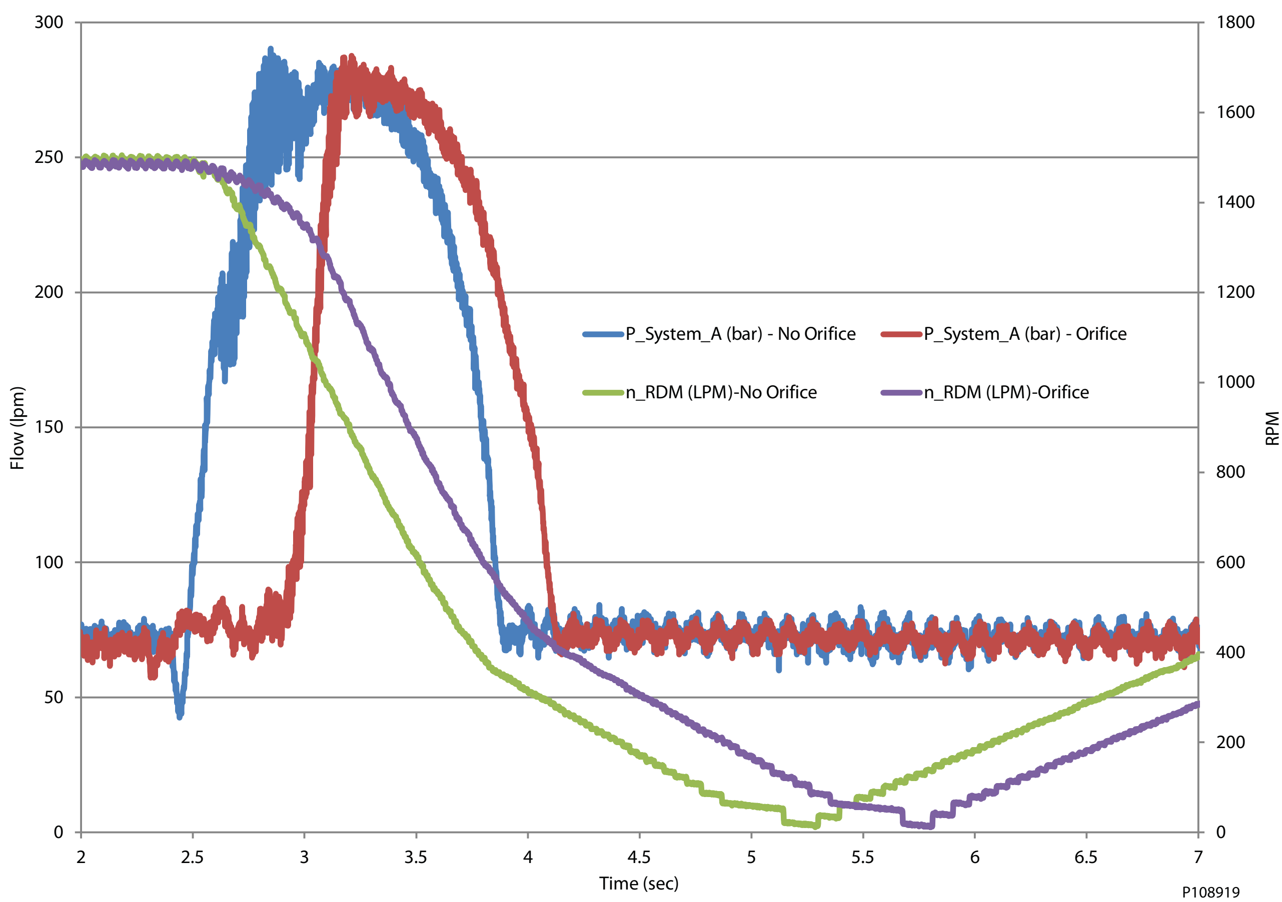
The “non-orifice” shift time of the motor with 12 bar or greater system pressure is approximately 350 msec. An orifice of 0.8mm (0.032”) in the supply position, or an orifice of 1.3mm (0.051”) between the control valve and servo piston, will approximately double this time. Slowing the motor’s shift rate will reduce, if not eliminate, the initial pressure spike during the shaft reversal and will allow the load to slow down some before the motor actually reverses into pumping mode. The pressure during the rest of the load deceleration will be similar to the “non-orifice” system interval. The addition of either orifice will result in a longer deceleration time interval.

The process of returning to forward is not affected by a supply orifice, but will be affected by an orifice between the control valve orifice and the servo piston. The total time to stop the fan in this direction is nearly the same but more time is spent at high pressure if an orifice after the control valve is used.

The preference would be to apply the motor without the orifice since this allows the motor itself to shift at lower system pressures. If high pressure spikes, or system instability, are seen during the forward to reverse step, the best option is introduce a supply orifice into the control circuit.

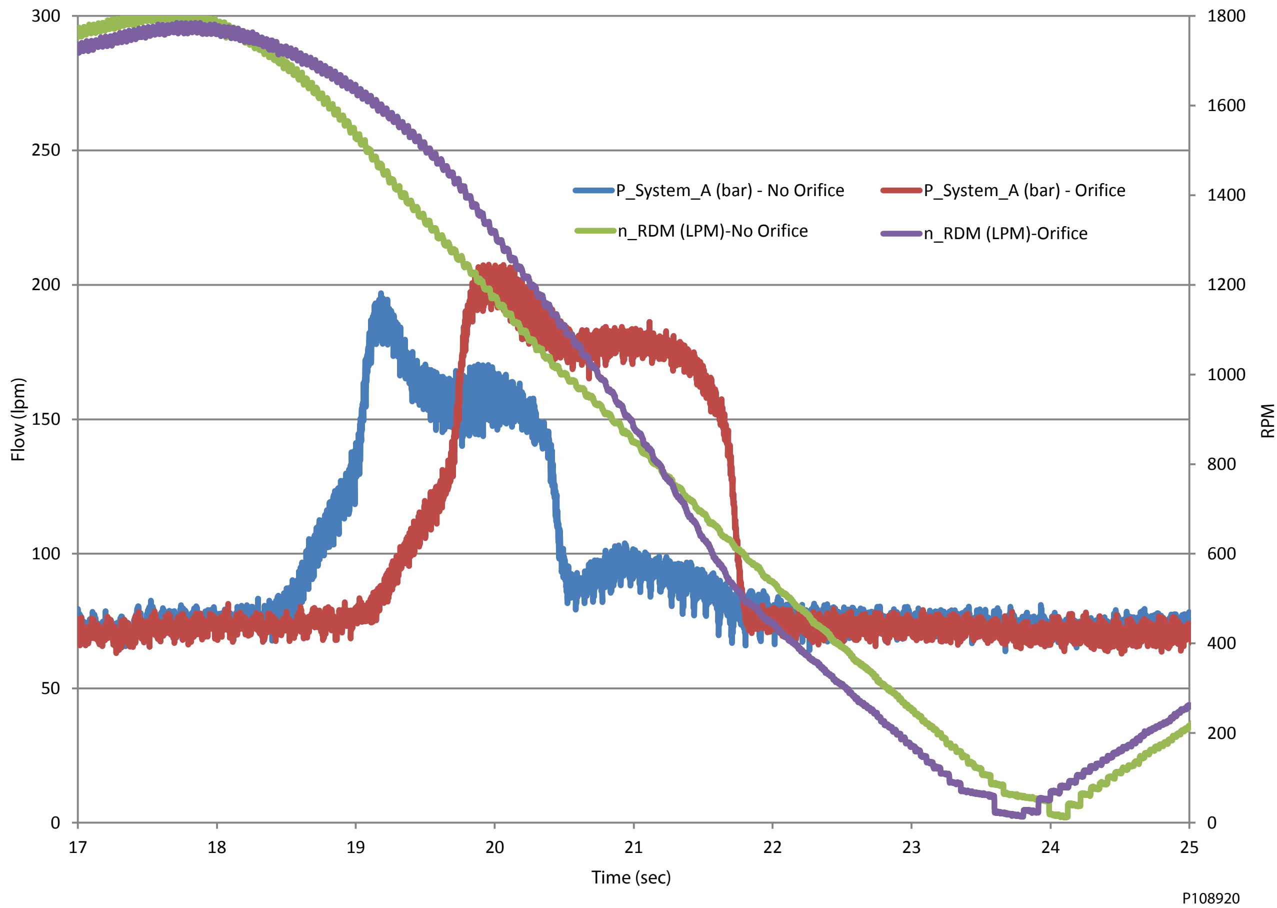
Appendix I-System Considerations for RDM Fan Drives

Control Orifice Impact Forward to Reverse



Appendix I-System Considerations for RDM Fan Drives

Control Orifice Impact Reverse to Forward



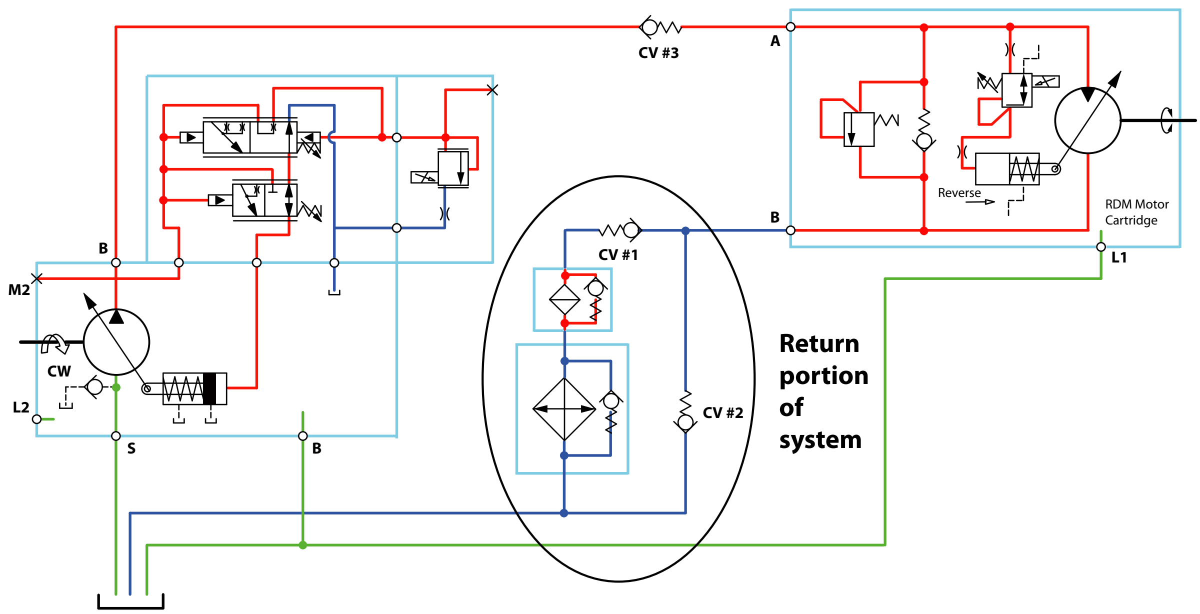
System Considerations

There are some system considerations that need to be taken into account when applying an RDM motor.

1. Anytime the RDM is used in a typical system, as shown in the schematic, a displacement reversal command will cause the RDM to turn into a pump until the load is stopped. This will cause oil to flow from the motor toward the pump (opposite the normal direction) with the amount of flow and length of time depending on system parameters such as the fan inertia, fan speed at command initiation, and reversing rate. When this happens, the motor (now operating as a pump) must be able to draw oil in through what is normally the return flow portion of the system.

Appendix I-System Considerations for RDM Fan Drives

RDM System Schematic



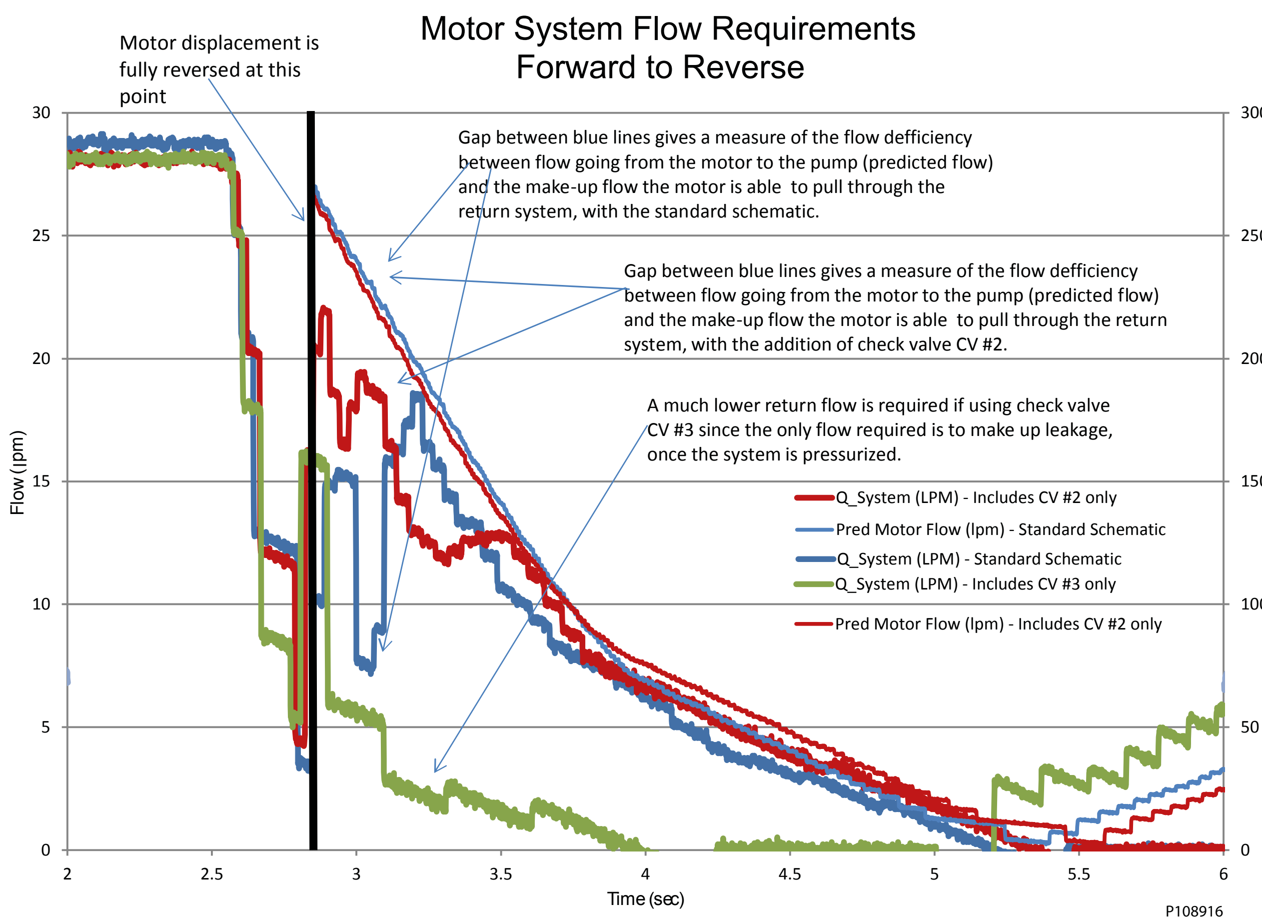
The amount of flow is a function of the motor displacement and speed so a system operating using the recommended reversing cycle will have only a very small amount of flow in the reverse direction, since the reversal starts at a much lower motor speed. A cycle with a high deceleration rate however, as shown in Control Orifice Impact graphs above, will require much greater amounts of flow, since the motor displacement is reversed at a much higher speed. Any filters, coolers, or other components that are in the return line could cause a restriction leading to cavitation in the motor. In addition to adding to any noise during the reversal, this could eventually lead to damage to the filter, heat exchanger, motor, or pump - causing reduced performance, or life, of the system.

One way to reduce this concern is to have a by-pass check valve in parallel to the filter and heat exchanger to allow free flow of oil into the motor when required. This is identified as CV #2 in the above schematic. This would also reduce the risk of this oil flow pulling contamination from the filter or damaging either of these components. This risk can be further minimized by adding a second check valve, shown as CV #1 in the schematic.

Motor Flow Requirements in Pumping Mode on page 73, below, shows an example in one system of the impact of such a by-pass valve. When comparing the “Standard Schematic” data to the “Includes CV#2 only” data, during the first 250 msec after the motor displacement is reversed, the system without the by-pass valve shows a much greater deficiency in make-up flow into the motor than did the system with the by-pass valve.

Appendix I-System Considerations for RDM Fan Drives

Motor Flow Requirements in Pumping Mode



2. The peak value, and duration, of any pressure spike in the high pressure line will be controlled by how quickly the load energy is put into the system and by the avenues available to the system to drain this energy out. The avenues in a typical open circuit system would be;
   - Pump and motor control leakage flow
   - Pump and motor kit leakage flow
   - Pump's ability to absorb oil by going over-center
   - Relief valves in the system

   The characteristics of each of the above items are discussed in more detail below. By looking at the energy in the load during the reversal initiation, the power level required to reverse the load in the desired time and the paths available for energy dissipation; the system designer can determine if any other actions might be required.
3. Another system consideration is the impact on the case drain system. During the time interval of the reversal, the output flow at the motor case drain is significantly higher than under normal conditions during the reversal process. The servo piston will force approximately 53 cc's (3.2 in^3) of oil into the motor case. Introducing 53 cc's of fluid in 350ms equates to a nearly instantaneous change in the average case flow rate of approximately 9 lpm (2.4 gpm).

   Also, in a system set-up like the schematic above, there may be a surge of oil through the pump control resulting in higher case flow leaving the pump. [Experience to date has not shown a problem with this on S45 pumps but this should be investigated during prototype testing to ensure there are no issues.]
4. Another system consideration is the impact on the system downstream of the motor outlet. During the motor reversal, the flow out of the motor will quickly reduce to zero (the motor's at zero

Appendix I-System Considerations for RDM Fan Drives

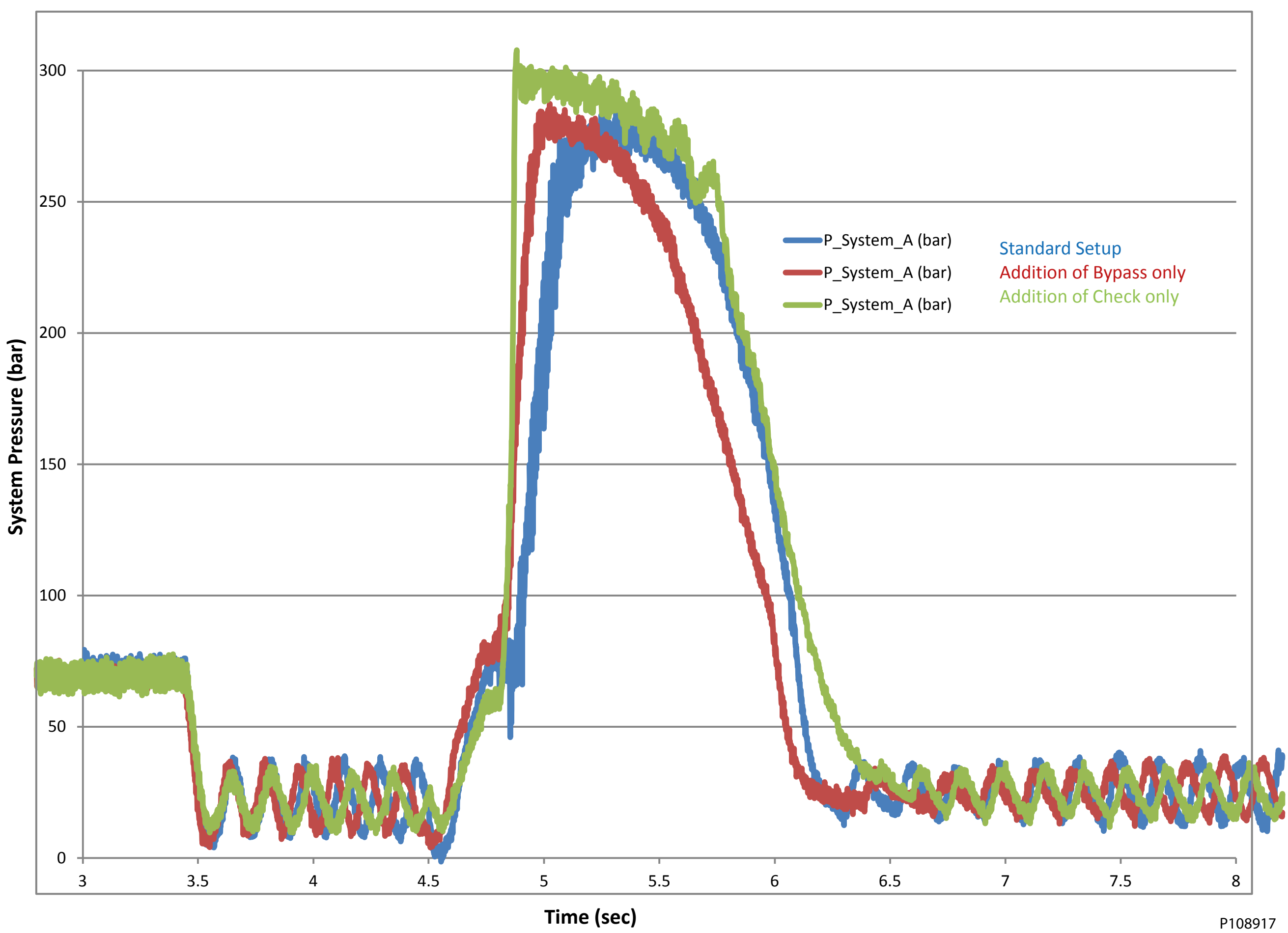
displacement) and then reverse direction (as discussed above). This has been shown to result in the motor outlet pressure cycling from normal - to a vacuum - to a high level spike. The main concern for the RDM itself is if the case-pressure rise (discussed directly above) and the outlet pressure drop - are too severe; such that added together they may cause a problem.

Laboratory tests following the recommended reversing cycle have shown that the RDM is more capable of handling a ("case minus return") delta pressure pulse higher than the catalog rating, but this value should be kept below 3 bar.

For example, a test was performed with a a S45 J open circuit pump, a K-Frame RDM motor, and a fan inertia of 2kgm^2. *System Pressure Response Characteristic* on page 74 below shows a typical system pressure response. The blue line, or "Standard Setup", would be the base system described in the schematic above. The red line, or "Addition of Bypass valve", reflects adding the bypass valve, CV #2, into the circuit. The system builds pressure more quickly and, as a consequence, slows the fan more quickly; this brings the pressure down sooner. [This is because the RDM is able to draw more oil in during its pumping mode, allowing the fan energy to be dissipated more quickly.]

The data in *System Pressure Response Characteristic* on page 74 shown by the green line represents the "addition of check valve, CV#3", (by itself) into the circuit between the pump and motor. It prevents the reverse flow from going into the pump. This system then utilizes only the motor leakage and integrated relief valve to dissipate the fan energy. As can be seen here, this results in a higher pressure and takes the longest time to reverse the fan.

System Pressure Response Characteristic



Appendix I-System Considerations for RDM Fan Drives

Zero RPM Motor Output

The RDM has the ability to reach and maintain a zero shaft speed condition. This could be beneficial for the purpose of reducing the time required for an engine to reach operating temperature, or for power savings - when cooling is not required.

A limited amount of tests have been performed in the laboratory to verify that this condition if is “possible”; but additional testing in the laboratory and on prototype vehicles under field conditions will be needed to verify whether a production solution on an individual vehicle system is feasible.

Warning

With regard to the swashplate bearings, at some point in the life of the motor, there is a potential that the motor will not return to the forward direction when the reverse command is removed. This is caused by excessive wear of the journal bearing and there are a number of factors that influence this life. Testing to date using the standard reversing cycle and schematic has shown have demonstrated a swashplate bearing life exceeding 60,000 reversals. This is believed to be satifactory for most applications.

Warning

With regard to the standard reversing cycle, a loss of control signal to the pump (due to a broken wire, corroded connection, etc.) would cause the pump to maintain maximum system pressure and motor speed. In this instance, the RDM will be able to reverse direction, but it may not be able to return to the forward direction, due to the high pressure and speed. As long as this does not become a standard operating mode, it will not damage the motor. Once the pressure and speed are reduced to their normal low pressure settings, the motor will return to the forward direction.

Reference Literature

Fan drive systems may consist of a variety of pump, motor, valve and control combinations. The product codes shown below are for components that have been designed specifically for fan drive systems. Refer to the literature listed below for product code information and specifications for other Danfoss components that may be utilized in a fan drive system.

Open Circuit Axial Piston Pumps

- *Series 45 Axial Piston Pumps* **520L0519**

Open/Closed Circuit Axial Piston Motors

- *L and K Frame Variable Motors* **520L0627**
- *Series 40 Axial Piston Motors* **520L0636**
- *Series 90 Axial Piston Motors* **520L0604**
- *H1B Axial Piston Motors* **11037153**

Controllers

- *Fan Drive Control Temperature Sensors* **BLN-95-9063**
- *Electronic Fan Drive Controller (FDC)* **11005336**
- *Electronic Fan Drive Controller Assembly (FDCA)* **11005337**
- *PLUS+1TM Controller Family* **520L0719**

System Guidelines

- *Design Guidelines for Hydraulic Fluid Cleanliness* **520L0467**
- *Design Guidelines for Hydraulic Fan Drive Systems* **520L0926**

Closed Circuit Axial Piston Pumps

- *H1 Axial Piston Pumps Basic Information* **11062168**
- *Series 42 Axial Piston Pumps* **11022637**
- H1 045/053 Tandem Axial Piston Pumps Technical Information **11063345**
- H1 045/053 Single Axial Piston Pumps Technical Information **11063344**
- H1 089/100 Single Axial Piston Pumps Technical Information **11069970**
- H1 069/078 Single Axial Piston Pumps Technical Information **11062169**
- H1 147/165 Single Axial Piston Pumps Technical Information **11063347**
- H1 210/250 Single Axial Piston Pumps Technical Information **L1208737**
- H1 115/130 Single Axial Piston Pumps Technical Information **11063346**

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