

(R) Aerospace - System Integration Factors That Affect Hydraulic Pump Life

RATIONALE

AIR1922 has been updated to Revision A for the following reasons:

Additional technical information has been introduced

The references called up in the document have been updated

Editorial changes have been made to improve the readability of the document

FOREWORD

The information provided in this Aerospace Information Report (AIR) is compiled from various sources and is based on the experience and judgment of engineering and operating personnel of hydraulic components and systems for aircraft.

This document is intended to be informative when considering hydraulic pump life and performance since there are no hard and fast rules for determining the optimum combination or arrangement of all the components in each hydraulic system.

However, the criteria to be followed and the operating parameters to be utilized when designing a hydraulic system must remain with the designer.

TABLE OF CONTENTS

1.	SCOPE.....	4
1.1	Field of Application.....	4
2.	REFERENCES.....	4
2.1	Applicable Documents.....	4
2.1.1	SAE Publications.....	4
2.1.2	RTCA Publications.....	5
2.1.3	US Government Publications.....	5
2.2	Related Publications.....	5
2.2.1	SAE Publications.....	5
2.3	Definitions.....	5
3.	PUMP INLET LINE.....	6
3.1	General.....	6
3.2	Inlet Pressure.....	6
3.3	Cavitation.....	7

SAE Technical Standards Board Rules provide that: "This report is published by SAE to advance the state of technical and engineering sciences. The use of this report is entirely voluntary, and its applicability and suitability for any particular use, including any patent infringement arising therefrom, is the sole responsibility of the user."

SAE reviews each technical report at least every five years at which time it may be reaffirmed, revised, or cancelled. SAE invites your written comments and suggestions.

Copyright © 2011 SAE International

All rights reserved. No part of this publication may be reproduced, stored in a retrieval system or transmitted, in any form or by any means, electronic, mechanical, photocopying, recording, or otherwise, without the prior written permission of SAE.

TO PLACE A DOCUMENT ORDER: Tel: 877-606-7323 (inside USA and Canada)
Tel: 724-776-4970 (outside USA)
Fax: 724-776-0790
Email: CustomerService@sae.org
http://www.sae.org

**SAE values your input. To provide feedback
on this Technical Report, please visit
<http://www.sae.org/technical/standards/AIR1922A>**

3.4	Air in Hydraulic Fluids	9
3.5	Boost Pumps	10
3.5.1	Integral Inlet Boost Stage in the Pump	10
3.5.2	Alternatives	11
3.6	Inlet Line - Sizing and Configuration	11
3.6.1	Pump Critical Inlet Pressure	11
3.6.2	Transient or Acceleration Pressure	12
3.6.3	Component Flow Losses	13
3.6.4	Line Friction Loss	14
3.6.5	Static Pressure Head	14
3.6.6	Example Calculation of Required Reservoir Pressure	33
3.6.7	Hydraulic Line Losses	36
4.	PUMP OUTLET LINE	37
4.1	Pressure Pulsations	38
4.1.1	General	38
4.1.2	Acceptable Level of Pressure Pulsations	39
4.1.3	Means to Reduce Pressure Pulsations	39
4.2	Hydraulic Pump Response Characteristics	40
4.3	Check Valve in Pump Outlet Line	42
4.4	Parallel Pump Installations	42
5.	PUMP CASE DRAIN LINE	42
5.1	General	42
5.2	Case Drain Pressure	42
5.2.1	Case Drain Scavenge Pumps	43
5.3	Case Drain Line - Direct to Reservoir and Return Line	43
5.4	Case Drain Flow Change	44
5.5	Parallel Pump Installations	44
5.6	Miscellaneous	44
5.6.1	Pump Connections	44
5.6.2	Quick Disconnect Couplings	44
5.6.3	Shut-off Valves	45
6.	MOUNTING PAD ACCESSORY DRIVE	45
6.1	General	45
6.2	Vibration	45
6.3	Pad Temperature	45
6.3.1	Soak Back	46
6.4	Pad Environment	46
6.5	Drive Alignment	46
6.6	Drive Splines	47
6.7	Torque	47
6.8	Speed	47
6.8.1	Low Speed	50
6.9	Miscellaneous	50
7.	FLUID PROPERTIES AND CLEANLINESS	50
7.1	Fluid Contamination	50
7.1.1	Fluid Contaminants	51
7.2	Filtration	51
7.2.1	System Filters	51
7.3	Fluid Monitoring	52
8.	NOTES	53

8.1	Revision Status	53
APPENDIX A	ASSOCIATED TECHNICAL DOCUMENTS	54
APPENDIX B	PUMP DRIVE SPLINE STANDARDS	55
FIGURE 1	TYPICAL CRITICAL INLET PRESSURE VALUES FOR INLINE PUMPS	8
FIGURE 2A	PRESSURE LOSS IN HYDRAULIC LINES - TURBULENT FLOW - MIL-PRF-5606, T = 180 °F (82 °C)	15
FIGURE 2B	PRESSURE LOSS IN HYDRAULIC LINES - TURBULENT FLOW - MIL-PRF-5606, T = 82 °C (180 °F)	16
FIGURE 3A	PRESSURE LOSS IN HYDRAULIC LINES - TURBULENT FLOW - MIL-PRF-87257, T = 180 °F (82 °C)	17
FIGURE 3B	PRESSURE LOSS IN HYDRAULIC LINES - TURBULENT FLOW - MIL-PRF-87257, T = 82 °C (180 °F)	18
FIGURE 4A	PRESSURE LOSS IN HYDRAULIC LINES - TURBULENT FLOW - MIL-PRF-83282, T = 180 °F (82 °C)	19
FIGURE 4B	PRESSURE LOSS IN HYDRAULIC LINES - TURBULENT FLOW - MIL-PRF-83282, T = 82 °C (180 °F)	20
FIGURE 5A	PRESSURE LOSS IN HYDRAULIC LINES - TURBULENT FLOW - AS1241, T = 180 °F (82 °C)	21
FIGURE 5B	PRESSURE LOSS IN HYDRAULIC LINES - TURBULENT FLOW - AS1241, T = 82 °C (180 °F)	22
FIGURE 6	EFFECT OF TEMPERATURE - TURBULENT FLOW	23
FIGURE 7A	PRESSURE LOSS IN HYDRAULIC LINES - LAMINAR FLOW - MIL-PRF-5606, T = 0 °F (-18 °C)	24
FIGURE 7B	FOR PRESSURE LOSS IN HYDRAULIC LINES - LAMINAR FLOW - MIL-PRF-5606, T = -18 °C (0 °F)	25
FIGURE 8A	PRESSURE LOSS IN HYDRAULIC LINES - LAMINAR FLOW - MIL-PRF-87257, T = 0 °F (-18 °C)	26
FIGURE 8B	PRESSURE LOSS IN HYDRAULIC LINES - LAMINAR FLOW - MIL-PRF-87257, T = -18 °C (0 °F)	27
FIGURE 9A	PRESSURE LOSS IN HYDRAULIC LINES - LAMINAR FLOW - MIL-PRF-83282, T = 0 °F (-18 °C)	28
FIGURE 9B	PRESSURE LOSS IN HYDRAULIC LINES - LAMINAR FLOW - MIL-PRF-83282, T = -18 °C (0 °F)	29
FIGURE 10A	PRESSURE LOSS IN HYDRAULIC LINES - LAMINAR FLOW - AS1241, T = 0 °F (-18 °C)	30
FIGURE 10B	PRESSURE LOSS IN HYDRAULIC LINES - LAMINAR FLOW - AS1241, T = -18 °C (0 °F)	31
FIGURE 11	EFFECT OF TEMPERATURE - LAMINAR FLOW	32
FIGURE 12	TYPICAL PRESSURE TRANSIENT FULL FLOW TO ZERO FLOW	41
FIGURE 13	TYPICAL PRESSURE TRANSIENT ZERO FLOW TO FULL FLOW	41
FIGURE 14A	MAXIMUM PUMP DISPLACEMENT VERSUS PUMP SPEED (IMPERIAL UNITS)	48
FIGURE 14B	MAXIMUM PUMP DISPLACEMENT VERSUS PUMP SPEED (METRIC UNITS)	49
TABLE 1	SUGGESTED IN-SERVICE FLUID LIMITS	52

1. SCOPE

This AIR presents the following factors that affect hydraulic pump life and performance:

- a. The need to supply hydraulic fluid at the correct pressure and quality to the pump inlet port
- b. Considerations for the pump output
- c. Factors to be considered for the pump case drain lines
- d. The mounting of the hydraulic pump
- e. Hydraulic fluid properties, including cleanliness

1.1 Field of Application

This document applies to pumps operating constantly or for long periods at quasi-constant speed or within narrow speed ranges, such as engine driven or electric motor driven pumps.

Wide speed range and intermittent duty cycle pumps, such as those used in Electrohydrostatic Actuators, are not covered by this document.

2. REFERENCES

2.1 Applicable Documents

The following publications form a part of this specification to the extent specified herein. The latest issue of SAE publications shall apply. The applicable issue of other publications shall be the issue in effect on the date of the purchase order. In the event of conflict between the text of this specification and references cited herein, the text of this specification takes precedence. Nothing in this specification, however, supersedes applicable laws and regulations unless a specific exemption has been obtained.

NOTE: Appendix A provides a list of documents that provide additional hydraulic system design information. It should be noted that some documents may be difficult to obtain.

2.1.1 SAE Publications

Available from SAE, 400 Commonwealth Drive, Warrendale, PA 15096-0001, Tel: 877-606-7323 (inside USA and Canada) or 724-776-4970 (outside USA), www.sae.org.

AIR810	Degradation Limits of Hydrocarbon-Based Hydraulic Fluids, MIL-H-5606, MIL-H-6083, MIL-H-83282, and MIL-H-46170 Used in Hydraulic Test Stands
AIR4543	Aerospace Hydraulics and Actuation Lessons Learned
AIR5277	Aerospace Fluid Power - Waste Reduction Practices for Used Phosphate Ester Aviation Hydraulic Fluid
AIR5829	Air in Aircraft Hydraulic Systems
AS595	Aerospace – Civil Type Variable Delivery, Pressure Compensated, Hydraulic Pump
AS1241	Fire Resistant Phosphate Ester Hydraulic Fluid for Aircraft
AS4059	Aerospace Cleanliness Classification for Hydraulic Fluids
AS19692	Pumps, Hydraulic, Variable Delivery, General Specification For

2.1.2 RTCA Publications

Available from Radio Technical Commission for Aeronautics, Inc., 1828 L Street, NW, Suite 805, Washington, DC 20036, Tel: 202-833-9339, www.rtca.org.

RTCA/DO-160 Environmental Conditions and Test Procedures for Airborne Equipment

2.1.3 US Government Publications

Available from Document Automation and Production Service (DAPS), Building 4/D, 700 Robbins Avenue, Philadelphia, PA 19111-5094. Tel: 215-697-6257, <http://assist.daps.dla.mil/quicksearch>.

MIL-STD-810 Environmental Test Methods and Engineering Guidelines

MIL-PRF-5606 Hydraulic Fluid Petroleum Base Aircraft, Missile and Ordnance

MIL-F-8815 Filter and Filter Elements, Fluid Pressure, Hydraulic Line, 15 Micron Absolute and 5 Micron Absolute, Type II Systems, General Specification For

MIL-PRF-83282 Hydraulic Fluid, Fire Resistant Synthetic Hydrocarbon Base, Aircraft

MIL-PRF-87257 Hydraulic Fluid, Fire Resistant Synthetic Hydrocarbon Base, Aircraft

2.2 Related Publications

The following publications are provided for information purposes only, and is not a required part of this SAE Aerospace document.

2.2.1 SAE Publications

Available from SAE International, 400 Commonwealth Drive, Warrendale, PA 15096-0001, Tel: 877-606-7323 (inside USA and Canada) or 724-776-4970 (outside USA), www.sae.org.

ARP4752 Aerospace - Design and Installation of Commercial Transport Aircraft Hydraulic Systems

AS5440 Hydraulic Systems, Aircraft, Design and Installation Requirements For

2.3 Definitions

DISSOLVED AIR: This is air that is physically in solution in the hydraulic fluid.

ENTRAINED AIR: This is air in the form of bubbles that are typically less than 0.04 inch (1 mm) in diameter that are dispersed in the hydraulic fluid.

FREE AIR: This is a pocket of air that is trapped in part of a hydraulic system

HYDRAULIC PUMP STABILITY: This is the freedom from persistent or quasi-persistent oscillation of the delivery control mechanism.

OUTGASSING PRESSURE: This is the pressure below which dissolved gas begins to separate from the fluid and becomes free or entrained.

RESPONSE TIME: This is the rate of change of displacement of the pump when subjected to a specific discharge circuit transient. On most pump models it is not possible to measure the pump displacement. Instead, a test can be carefully designed wherein it is possible to deduce the change in displacement of the pump by the inspection of the pump discharge pressure. This measurement is called "Response Time" as it is a dynamic characteristic of a pump operating in a specified discharge circuit. It is not strictly an attribute of the pump. In such a test, the response time is the time interval between the instant when an increase (or decrease) in discharge pressure change initiates; and the subsequent instant when the discharge pressure reaches its first maximum (or minimum) value.

SATURATION PRESSURE: This is the static pressure below which there occurs a net migration of molecules (normally atmospheric) to the gaseous state (outgassing). The steady-state saturation pressure of hydraulic fluid at rest in an open container is the local atmospheric pressure. The saturation pressure in a system may not be homogeneous and may change over time.

VAPOR PRESSURE: This is the pressure below which the fluid changes phase and forms vapor bubbles.

3. PUMP INLET LINE

3.1 General

Any positive displacement pump designed to flow a liquid requires some minimum pressure at its inlet port to produce satisfactory performance. The magnitude of this minimum pressure (expressed in terms of absolute pressure such as psia (kPa abs)) is a function of the pump's design, shaft speed, duty cycle and the viscosity of the liquid being pumped. If the pump is mounted inside an open reservoir of liquid, the inlet pressure will be atmospheric plus the "head" of liquid above the pump. If such an installation does not provide the necessary inlet pressure, the reservoir could be closed and pressurized by some means to the pressure required by the pump for the particular set of operating conditions.

For numerous reasons, it is often impractical to mount the pump inside a reservoir, thus the need for an inlet line to connect the source of liquid with the pump inlet port. Furthermore, in practice the medium is not a pure liquid but a mixture of fluids, as there is always some air mixed in it. To the extent that the inlet line affects the net inlet pressure and quality of the fluid available at the pump, it has a significant effect on the pump's performance and life. (The term "quality" here refers to the form of air mixed in the liquid.)

The primary factors to be considered so as to ensure that the inlet line is optimized for supplying fluid to the pump are:

- Its cross-section
- Length, routing
- Restrictions
- Air tightness

3.2 Inlet Pressure

Despite the many improvements made to hydraulic fluids (e.g., lubricity, shear strength, viscosity stability, etc.), one physical characteristic cannot be changed, and that is the complete lack of tensile strength of the fluid. It is, therefore, impossible to move a quantity of liquid, larger than a drop, by pulling on it and to move a fluid it must be pushed, that is, propelled by pressure behind it – even to the inlet of a pump.

To obtain satisfactory performance and life from a positive displacement pump, its internal pumping elements must remain filled with fluid, ideally as close as possible to a pure liquid. To achieve this, not only must free air be avoided, but also the fluid environment must remain above the fluid's outgassing pressure and especially above its vapor pressure. For the fluids of interest, even at their maximum system operating temperature, the vapor pressure is typically below 1.0 psia (6.9 kPa abs), which may sound too low to be of practical concern. However, this low pressure can easily occur at the valving surfaces inside a pump operating at high speeds or during flow transients in long inlet lines.

NOTE: Airframe inertial accelerations also can act on the fluid mass and that the momentum of flowing fluid are also factors.

Between the pump inlet port and the actual internal pumping elements, there are, of necessity, fluid passages of discrete dimensions. In addition, there is a valving mechanism to separate the inlet area from the high pressure discharge area. These generate a restriction to fluid flow, hence a pressure drop. Consequently, to maintain close to saturation pressure in the pumping elements, the pressure at the inlet port must be somewhat higher. The resulting pressure for practical purposes is designated as the "Critical Inlet Pressure of the Pump".

The magnitude of the critical inlet pressure will vary as a function of pump design, operating conditions and fluid characteristics. It should be noted that at low temperatures (typically between 32 °F to 64 °F (0 °C to 16 °C)), the fluid flow tends to become laminar, thus increasing the critical inlet pressure due to fluid viscosity effects within the pump.

Typical critical inlet pressure (P_{crit}) requirements are shown in Figure 1 for various pump sizes. The critical inlet pressure for a specific pump should be readily available from the pump manufacturers.

Operation below this critical inlet pressure will cause pump cavitation with subsequent reduction in outlet flow, increased pressure ripple, and result in severe damage to the pump. The amount of damage depends on how much the inlet pressure was below the critical value, and for how long. While periods of only a few milliseconds will cause only limited damage, the destructiveness is cumulative and can manifest itself as serious damage in less than a hundred hours.

It is also important to remember that the measured pressure at the pump inlet may appear adequate even though the fluid contains substantial amounts of undissolved air. In such cases, pump damage will still occur.

3.3 Cavitation

If the inlet fluid contains entrained (undissolved) gas, typically air, or if the inlet pressure of the fluid is allowed to drop below its outgassing pressure so that dissolved air comes out of solution or worse yet, below its vapor pressure, there will be bubbles. After these enter the pumping elements and are then compressed to pump discharge pressure, they will collapse violently and generate a shockwave. In the case of piston pumps, this violent release of localized energy will manifest itself as cavitation erosion in such areas as:

- Cylinder blocks/cylinder barrels
- Valving plates/port plates
- Shoes/slippers
- Shoe bearing/wear plates

The occurrence of cavitation is clearly noticeable by its characteristic noise and line vibration. However, in the case of pressure transients, the noise and vibration may be masked by other factors. Damage to the pumping elements as a result of cavitation consists of small particles of material being removed from their surfaces and subsequently carried by the fluid through the discharge and case drain ports into the system. Since there is material removed, the damage is cumulative.

The material removal can cause changes to pump timing and hydrostatic balance, which in turn can cause premature wear of the sliding surfaces. Additionally, the wear debris in suspension may score other hydrodynamic sliding surfaces thereby creating more debris. Most of the contaminants will then circulate through the pump bearings precipitating their premature failure.

It is therefore important that pump cavitation is avoided at all times, including during transient conditions. The design of the system and components should ensure that the required inlet pressure on the fluid at the pump is always maintained.

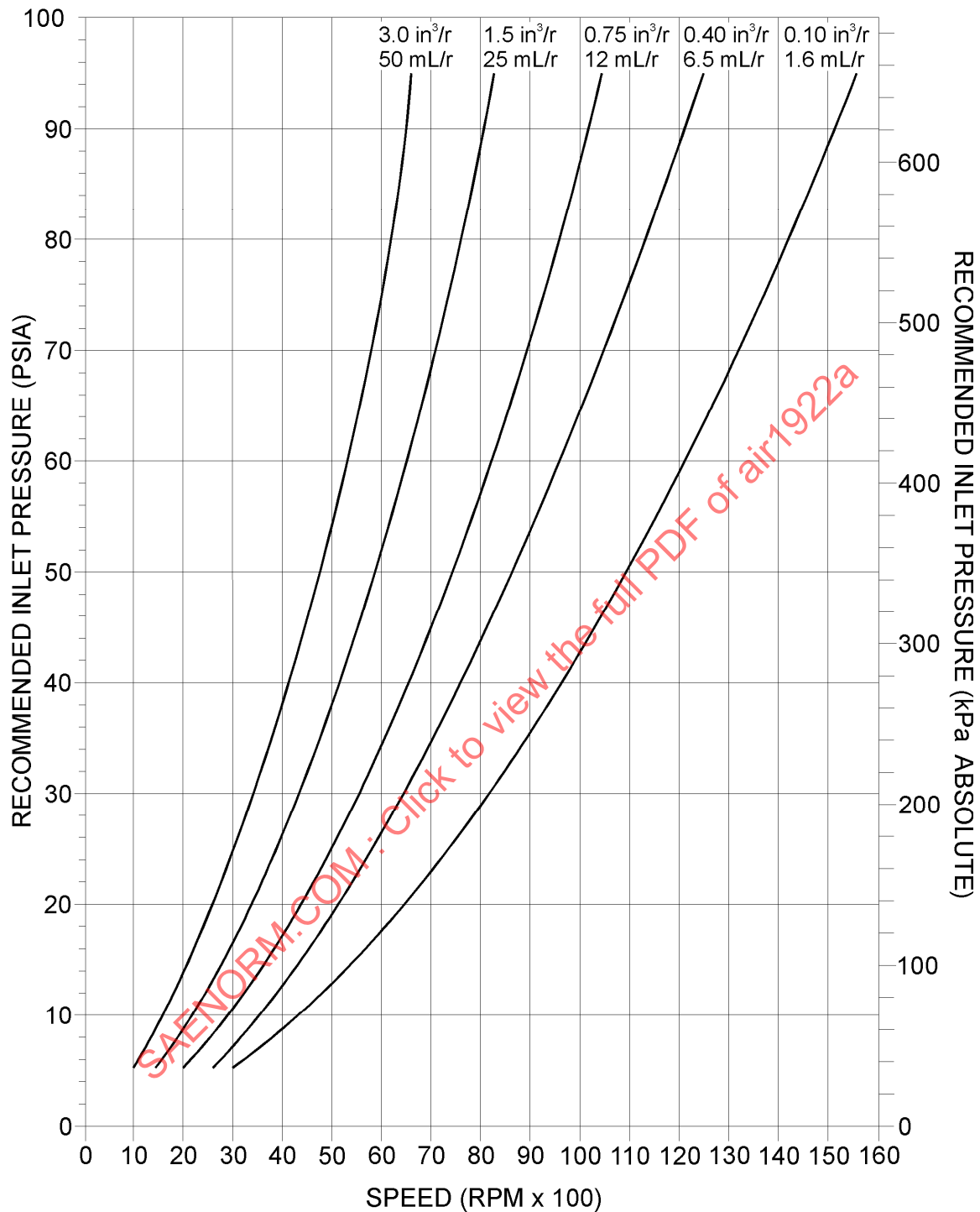


FIGURE 1 - TYPICAL CRITICAL INLET PRESSURE VALUES FOR INLINE PUMPS

- NOTES: 1. Consult the hydraulic pump supplier for the definitive critical inlet pressure for a specific pump application.
2. The inlet pressures shown in Figure 1 are for pumps operating with MIL-PRF-5606, MIL-PRF-83282 and MIL-PRF-87257 hydraulic fluids at nominal fluid temperatures. An increase of 30% is recommended for operation with AS1241 hydraulic fluids to allow for their increased density compared to the MIL-PRF fluids.

3.4 Air in Hydraulic Fluids

The hydraulic system can contain air in three forms:

- a. Dissolved air
- b. Entrained air
- c. Free air

The first two were commented on as they affect cavitation, and will be cited again in relation to inlet line configuration.

Extensive articles have been written on air in hydraulic fluids, including experience with quantitative amounts of dissolved and entrained air.

Free air in a hydraulic system can also cause extensive damage to pumps as well as other components. It does so primarily by interrupting the lubricating fluid film in critical areas, hence allowing metal-to-metal contact. Aside from pump damage, free air (often manifested as foam) can have serious operational effects such as:

- System instability
- Increased pressure ripple
- The loss of pump prime
- The inability of the pump to maintain flow and pressure
- Damaging the hydraulic fluid over time because when free air is compressed to design operating pressure, will also create a very tiny, but extremely hot, pocket of gas.

The presence of free air is often due to inadequate bleeding of the system; however, poorly thought-out design details can make adequate bleeding next to impossible.

System design details that will help avoid serious air problems include:

- a. Ensure that the pump case drain port is located in the upward direction
- b. Avoid non-uniformity in line size and sharp bends in the pump inlet line
- c. As hoses generally cause a larger pressure drop than tubing, they should be used with caution
- d. Design tubing routes to prevent air, water, and particulate entrapment and avoid vertical loops in the inlet line as this increases the risk of having air traps which make pump priming and system bleeding very difficult and sometimes impossible during adverse conditions.
- e. Ensure that hydraulic controls elevated above the reservoir can be adequately bled during ground maintenance activities
- f. Install bleed valves at high points in the hydraulic system to remove entrapped air

NOTE: Bleed valves are normally installed where there is an air trap combined with low pressures and low flow rates.

- g. Minimize the number of fittings in pump inlet lines. While these may not leak hydraulic fluid, they can ingest air

h. With bootstrap reservoirs:

1. Provide low piston friction to reduce the vacuum effect in the inlet line as the hydraulic system cools down
2. Provide sufficient auxiliary force (i.e., spring, gas pressure, etc.) to retain the reservoir piston against the fluid
3. Provide automatic bleed valves permanently mounted at the highest point of the reservoir to bleed free air
4. Provide an in-system air eliminator to reduce dissolved air to below saturation conditions to further reduce the potential of free air
5. Assure components do not include restrictors that cause air formation as discussed in AIR5829 and AIR4543.

i. With air-pressurized reservoirs:

1. The use of a hydraulic fuse (by-pass valve) between the pump discharge and return or inlet may be desirable for the start-up condition
2. Use an air to oil separator in the reservoir

NOTE: If a separator is used, then the bootstrap reservoir requirements (h), 3,4 and 5 apply.

3. The use of unseparated, air-pressurized reservoirs can significantly increase the dissolved air content of the system which can lead to out-gassing in the pump suction line. In this arrangement, it may be advantageous to include an integral inlet boost stage in the pump (ref 3.5.1).

3.5 Boost Pumps

3.5.1 Integral Inlet Boost Stage in the Pump

When the inlet line is compromised, or reservoir pressure too low, the resulting inlet pressure may be too low for the positive displacement pump. In that case, the pump manufacturer may incorporate an integral inlet boost section, typically an impeller, into the pump.

The pressure boost capabilities of typical, well designed centrifugal impellers are in the 20 to 40 psi (138 to 276 kPa) range, depending on their size and operating speed, as well as the flow rate, fluid and temperature. When all these parameters are in their optimal combination, the critical inlet pressure of the pump combination can be reduced to the 5 to 10 psia (34.5 to 69 kPa abs) range. Effective performance without cavitation below approximately 5 psia (34.5 kPa abs) inlet pressure may require the addition of inducers or injectors, in addition to an impeller.

Integral boost mechanisms in the pump can be very effective and have the advantage of dissolving some of the entrained air, but add considerable size, weight and cost to the pump. Their design is very critical and maximum effectiveness is in a relatively narrow performance band in terms of speed and temperature, as compared to that of the positive displacement pump. In many situations, the total system would be better served by improving the inlet line (reducing its pressure drop) than by tolerating an inferior inlet line with the more complex two-stage or even three-stage pump combinations.

3.5.2 Alternatives

The following should be considered if it is not feasible or desirable to introduce an integral inlet boost section into the pump:

- Incorporate a centrifugal boost pump near the reservoir produces one of the best inlet conditions for the main pump as it minimizes the side effects of the air in the fluid
- Install a charged accumulator installed in close proximity to the pump inlet port if a poor inlet condition occurs only during transients
- To allow or specify a slower pump response when high flows are required from large displacement pumps with marginal inlet pressure
- Fittings and quick disconnects installed close to or in the pump inlet port are to have a minimal pressure drop. (This is to prevent air coming out of solution at the pump inlet).

3.6 Inlet Line - Sizing and Configuration

The design of the pump inlet line, reservoir location, and reservoir pressurization are major system elements affecting hydraulic pump life. Once the basic needs of pump inlet pressure and the inevitable consequences of air and cavitation are understood, attention to the components in the pump inlet line becomes more meaningful.

At the reservoir end, sufficient pressure must be available to push the fluid through the inlet line such that, at the pump inlet port, the fluid pressure is still above the pump's critical inlet pressure under the worst commonly encountered conditions.

To put these conditions in perspective, it must be remembered that the response requirements of many hydraulic systems demand that a variable volume pump responds from its zero flow "dead headed" condition to full flow output in 50 to 200 ms. While most pumps have little difficulty in providing such response, it must be remembered that "you cannot get out what you do not put in". Therefore, the inlet pressurization must be such that the entire mass of fluid in the inlet line will also be accelerated in 50 to 200 ms and still remain above the critical inlet pressure at the pump inlet port.

It follows then that reservoir pressure should be determined by the following elements:

$$P_{\text{resv}} = P_{\text{crit}} + \Delta P_{\text{acc}} + \Delta P_{\text{com}} + \Delta P_{\text{lin}} + \Delta P_{\text{sta}} \quad (\text{Eq. 1})$$

Where:

P_{resv} = Reservoir pressure

P_{crit} = Pump critical inlet pressure

ΔP_{acc} = Pressure to accelerate fluid mass in pump response time

ΔP_{com} = Pressure loss of inlet line components at maximum flow

ΔP_{lin} = Frictional pressure loss in inlet line and hose assembly at maximum flow

ΔP_{sta} = Static pressure head

3.6.1 Pump Critical Inlet Pressure

This is discussed in section 3.2.

3.6.2 Transient or Acceleration Pressure

The transient or acceleration pressure (ΔP_{acc}) is that pressure necessary to accelerate the fluid mass in the inlet line to the maximum required flow within the response time of the pump. The pressure can be determined from the basic equation for acceleration force ($F = ma$) as follows:

$$m = \rho AL \text{ (fluid density x volume)} \quad (\text{Eq. 2})$$

$$a = \frac{dv}{dt} \text{ (time rate of change of velocity)} \quad (\text{Eq. 3})$$

$$F = \rho AL \times \frac{dv}{dt} = \rho AL \times \frac{\frac{dQ}{A}}{dt} = \rho L \times \frac{dQ}{dt} \quad (\text{Eq. 4})$$

The acceleration pressure needed to produce this force is:

$$\Delta P_{acc} = \frac{F}{A} = \frac{\rho L}{\pi/4 \times D^2} \times \frac{dQ}{dt} \quad (\text{Eq. 5})$$

where:

ΔP_{acc} = Pressure required to accelerate, psi (kPa)

dQ = Maximum flow change (gal/min), gpm (L/min)

dt = Pump response (seconds), s

L = Inlet line length, ft (m)

D = Tube inside diameter, in (mm)

ρ = Mass density of the fluid, lb_m/ft³ (kg/L)

A = Tube inside cross-section area, in² (mm²)

When converted to a form that utilizes the commonly used engineering units, and assuming a linear pump response, the foregoing formula appears as follows:

$$\Delta P_{acc} = 88.1 \times 10^{-6} \frac{\rho L}{D^2} \times \frac{dQ}{dt} \text{ (English units)} \quad (\text{Eq. 6})$$

$$\Delta P_{acc} = 21.2 \frac{\rho L}{D^2} \times \frac{dQ}{dt} \text{ (metric units)}$$

3.6.2.1 Allowance for Fluid Acceleration Non-Uniformity

If the non-uniformity of the fluid acceleration due to wave effects is taken into account, then the transient pressure term ΔP_{acc} , is oscillatory, with its average value given by Equation 5, but with its maximum value given by:

$$\Delta P_{acc} = ((\rho \cdot c) \times (\Delta Q/A)) \quad (\text{Eq. 7})$$

where:

c = wave speed in fluid = $(\beta/\rho)^{0.5}$

β = adiabatic bulk modulus, psi (kPa)

ΔQ = flow change at pump during time $2\Delta t$, in³/sec

Δt = time for the pressure wave to travel distance L from pump to the reservoir = L/c

If dQ/dt is constant over the time $2\Delta t$, then ΔQ is equal to $2\Delta t \cdot dQ/dt$ or $2(L/c) \cdot dQ/dt$, so that substituting its value in Equation 7, one gets:

$$\text{Maximum value of } \Delta P_{acc} = 2 \rho L (dQ/dt) / A \quad (\text{Eq. 8})$$

When converted to a form that utilizes the commonly used engineering units, and assuming a linear pump response, the foregoing formula appears as follows:

$$\Delta P_{acc} = 176.2 \times 10^{-6} \frac{\rho L}{D^2} \frac{dQ}{dt} \quad (\text{English units}) \quad (\text{Eq. 9})$$

$$\Delta P_{acc} = 42.4 \frac{\rho L}{D^2} \times \frac{dQ}{dt} \quad (\text{metric units})$$

It should be noted that the derivation of equation 6b assumes that the fluid pressure stays high enough for the air in fluid to stay in solution and for no fluid turning from liquid into vapor. Also, if part of the inlet line is a tube and part of it is a hose, then the equations for the maximum value of ΔP_{acc} , become more complex and the use of a dynamic analysis program may be required.

3.6.3 Component Flow Losses

Component flow losses (P_{com}) are those generated by such typical items as a shutoff valve, check valves, filters, and fittings. All of these should be avoided where possible, and certainly minimized in their effect on the pressure drop in the inlet line. The pressure drop at rated flow for each of these components is obtained from their manufacturers and treated cumulatively for a total component pressure loss value.

A less obvious effect of inlet component restriction and the resultant pressure drop is the possibility of outgassing the fluid; that is, dissolved air coming out of solution. This can be thought of as a local phenomenon. Provided there is enough pressure left and sufficient distance between the component and the pump inlet, the air can go back into solution downstream prior to reaching the pump. It is, therefore, desirable to keep all inlet line components, including quick disconnect couplings, as far away as possible from the pump to give the fluid a chance to recover from the outgassing phenomenon.

A special note of caution is given for systems with an inlet line pressure below atmospheric pressure; some fittings will seal pressure adequately from the inside, but not necessarily seal as well from the outside. In such cases, air will leak into the inlet line.

3.6.4 Line Friction Loss

The next element in the sum of inlet pressure considerations is the fluid friction loss (ΔP_{lin}) in the line (including hoses) itself. The pressure drop in a line for maximum steady state flow can be calculated by the standard formulas used for turbulent or laminar flow conditions.

Paragraph 3.6.7 provides the method and formulas to determine the Reynolds number for laminar or turbulent flow conditions.

Pressure loss data for various pipe diameters and hydraulic fluids are provided as follows:

Figures 2a to 5b provide turbulent flow pressure loss data with the fluid temperature at 180 °F/82 °C

Figure 6 provides a pressure loss correction factor to enable the turbulent flow pressure losses to be estimated at other temperatures

Figures 7a to 10b provide laminar flow pressure loss data with the fluid temperature at 0 °F/-18 °C

Figure 11 provides a pressure loss correction factor to enable the laminar flow pressure losses to be estimated at other temperatures

NOTES:

1. Turbulent flow - for pressurized lines, increase the basic pressure loss by 3% per 1000 psi pressure
2. Laminar flow - for pressurized lines, increase the basic pressure loss by 15% per 1000 psi pressure.

It is recommended that the maximum pump flow at maximum application speed (not the maximum demand on the system) be used to calculate the steady state line loss. Line losses should be determined at the lowest temperature at which full system operation is required. This temperature is often -20 °F (-29 °C) for military and commercial aircraft applications. Fast response aircraft or missile systems may require full system operation at lower temperatures. Normally, an aircraft system is allowed time to warm to the -20 °F (-29 °C) operating temperature prior to requiring large flow demands from the pump.

3.6.5 Static Pressure Head

If the pump height is more than a few inches or centimeters above that of the reservoir, the static pressure head (ΔP_{sta}) must be added to the required reservoir pressure.

$$\Delta P_{\text{sta}} = .0007 \rho h_1 \text{ (English units)} \quad (\text{Eq. 10})$$

where:

ρ = Density of the fluid in lb_m/ft^3

h_1 = Height difference in feet

or

$$\Delta P_{\text{sta}} = 9.8 \rho h_1 \text{ (metric units)}$$

where:

ρ = Density of the fluid in kg/L

h_1 = Height difference in meters

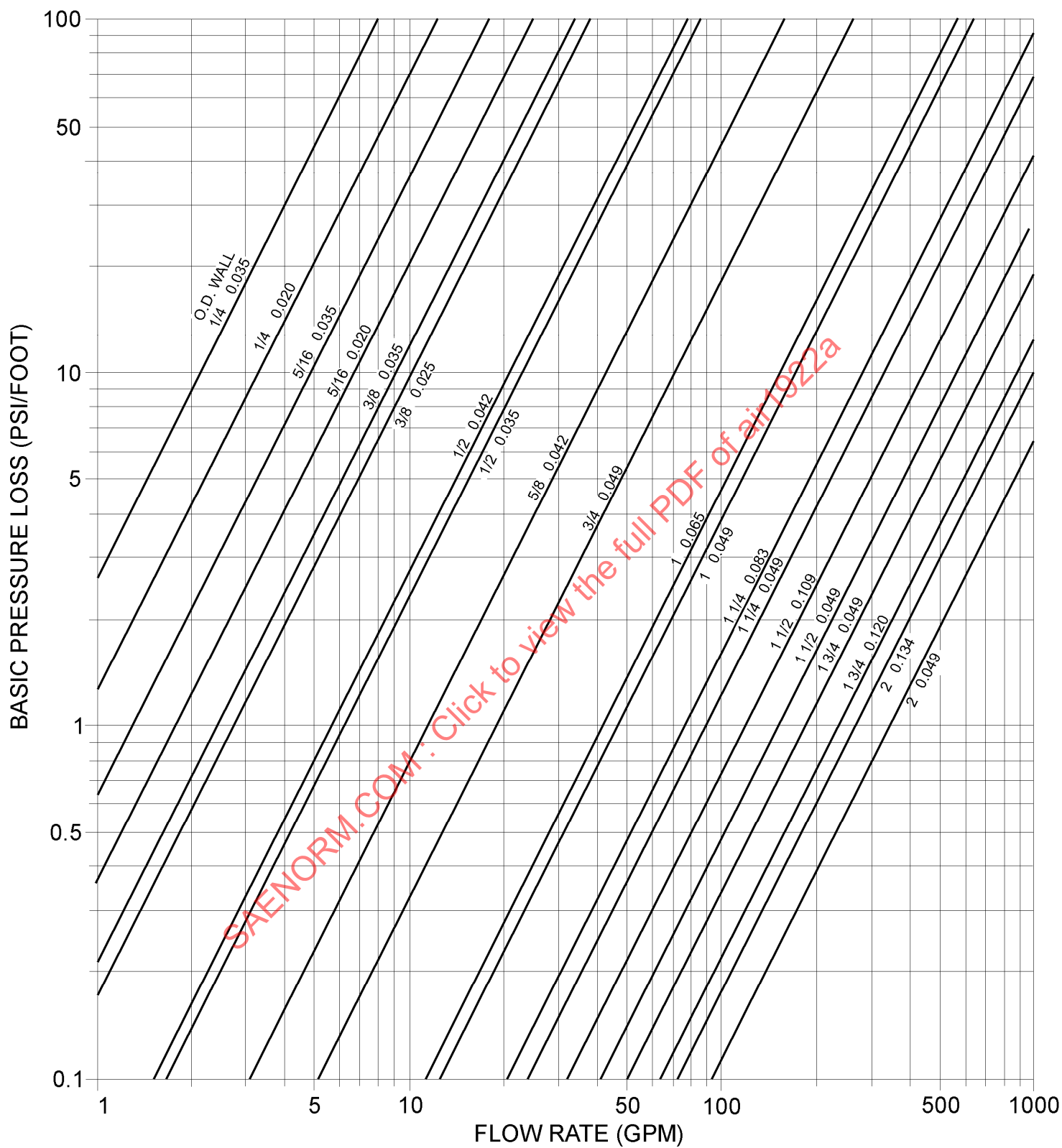


FIGURE 2A - PRESSURE LOSS IN HYDRAULIC LINES - TURBULENT FLOW - MIL-PRF-5606, T = 180 °F (82 °C)

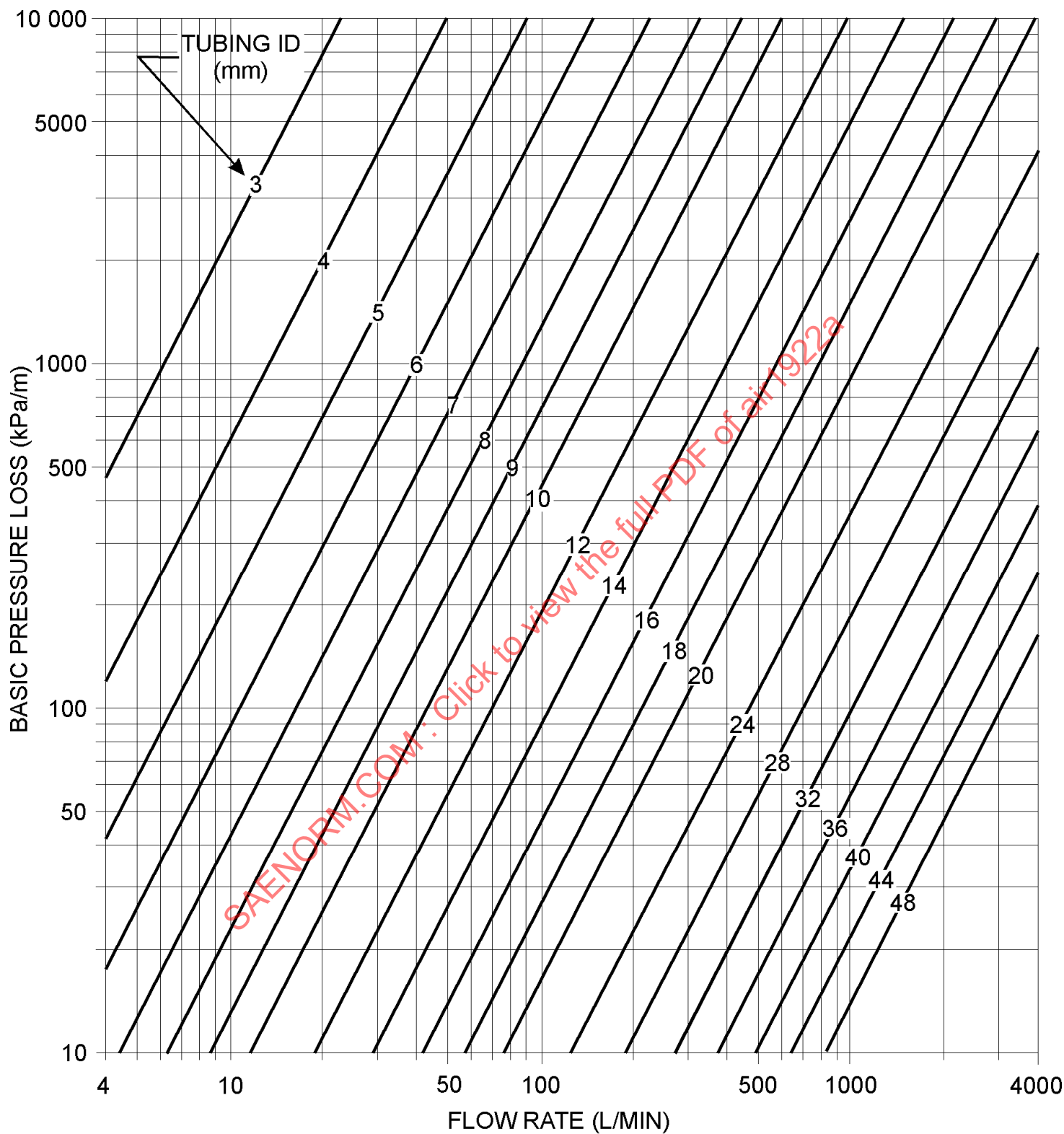


FIGURE 2B - PRESSURE LOSS IN HYDRAULIC LINES - TURBULENT FLOW - MIL-PRF-5606, T= 82 °C (180 °F)

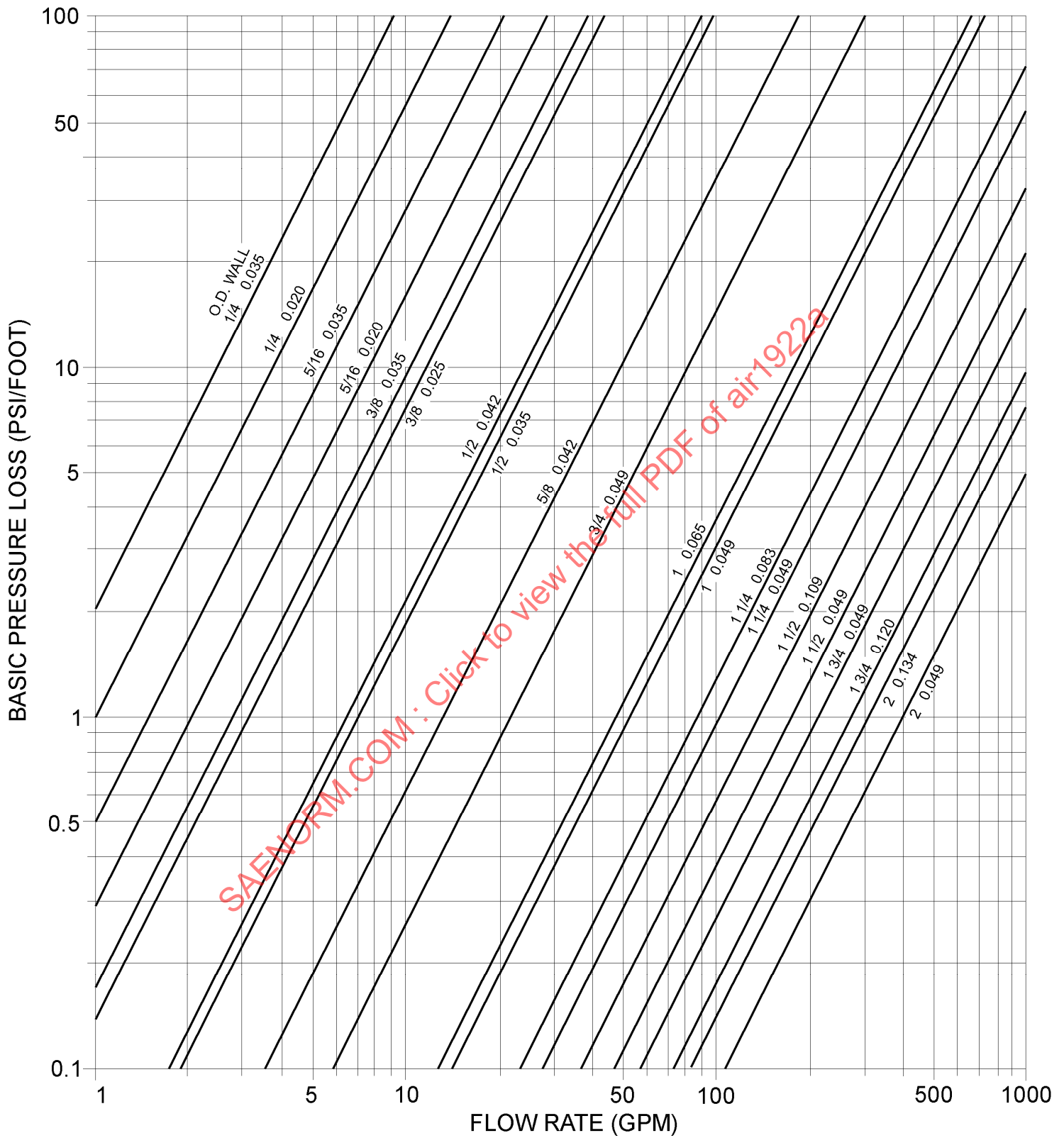


FIGURE 3A - PRESSURE LOSS IN HYDRAULIC LINES - TURBULENT FLOW - MIL-PRF-87257, T = 180 °F (82 °C)

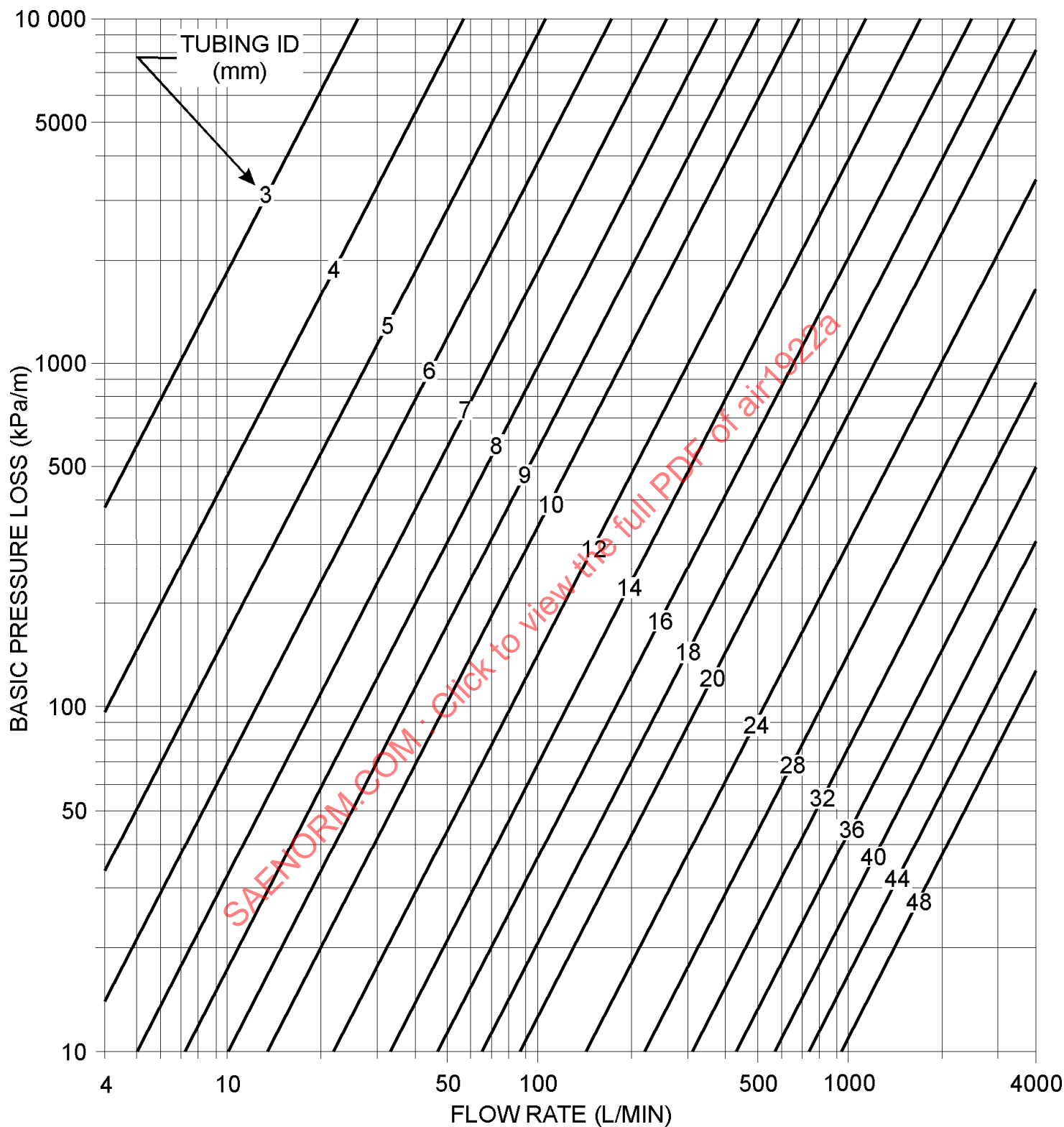


FIGURE 3B - PRESSURE LOSS IN HYDRAULIC LINES - TURBULENT FLOW - MIL-PRF-87257, T= 82 °C (180 °F)

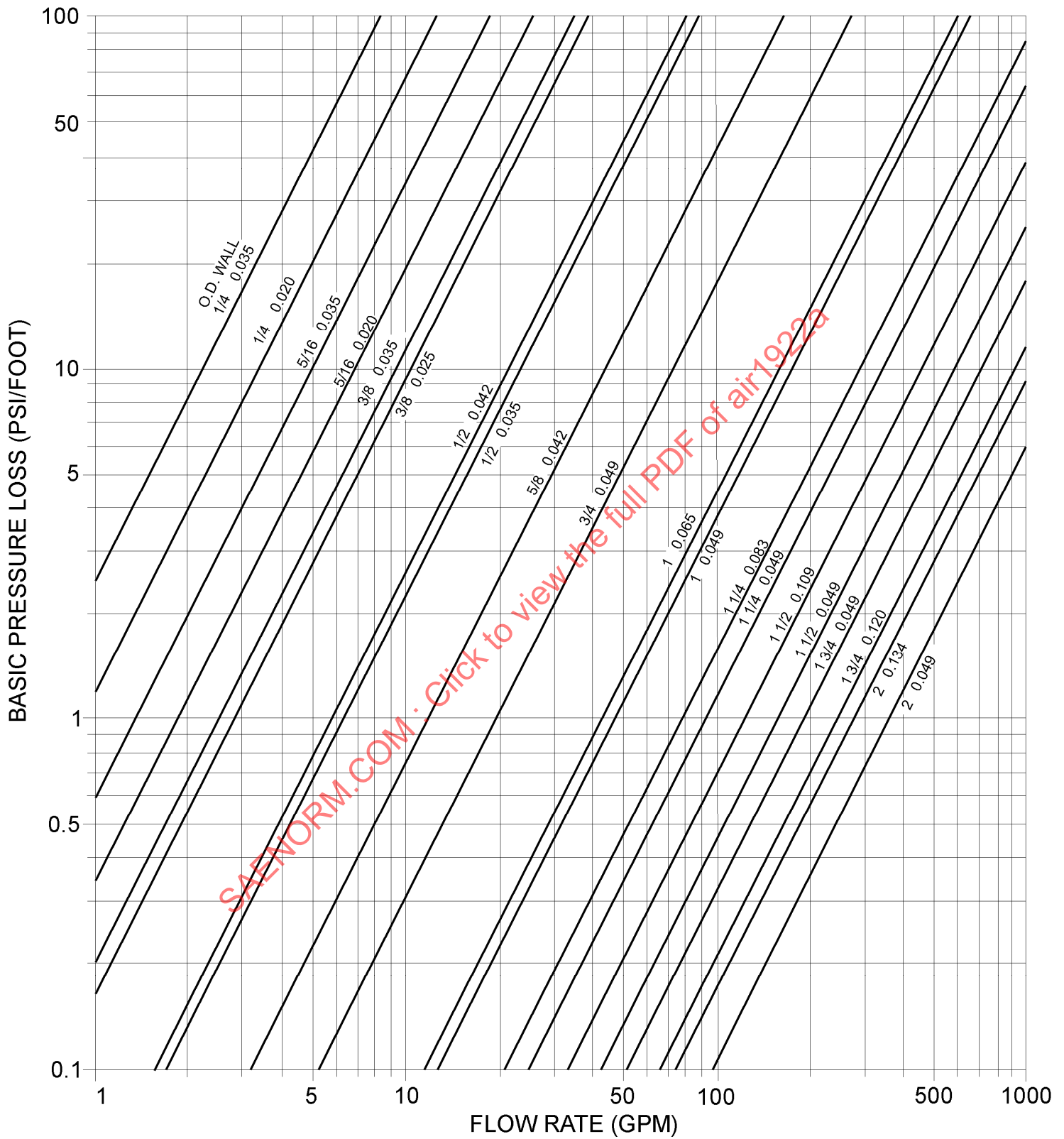


FIGURE 4A - PRESSURE LOSS IN HYDRAULIC LINES - TURBULENT FLOW - MIL-PRF-83282, T = 180 °F (82 °C)

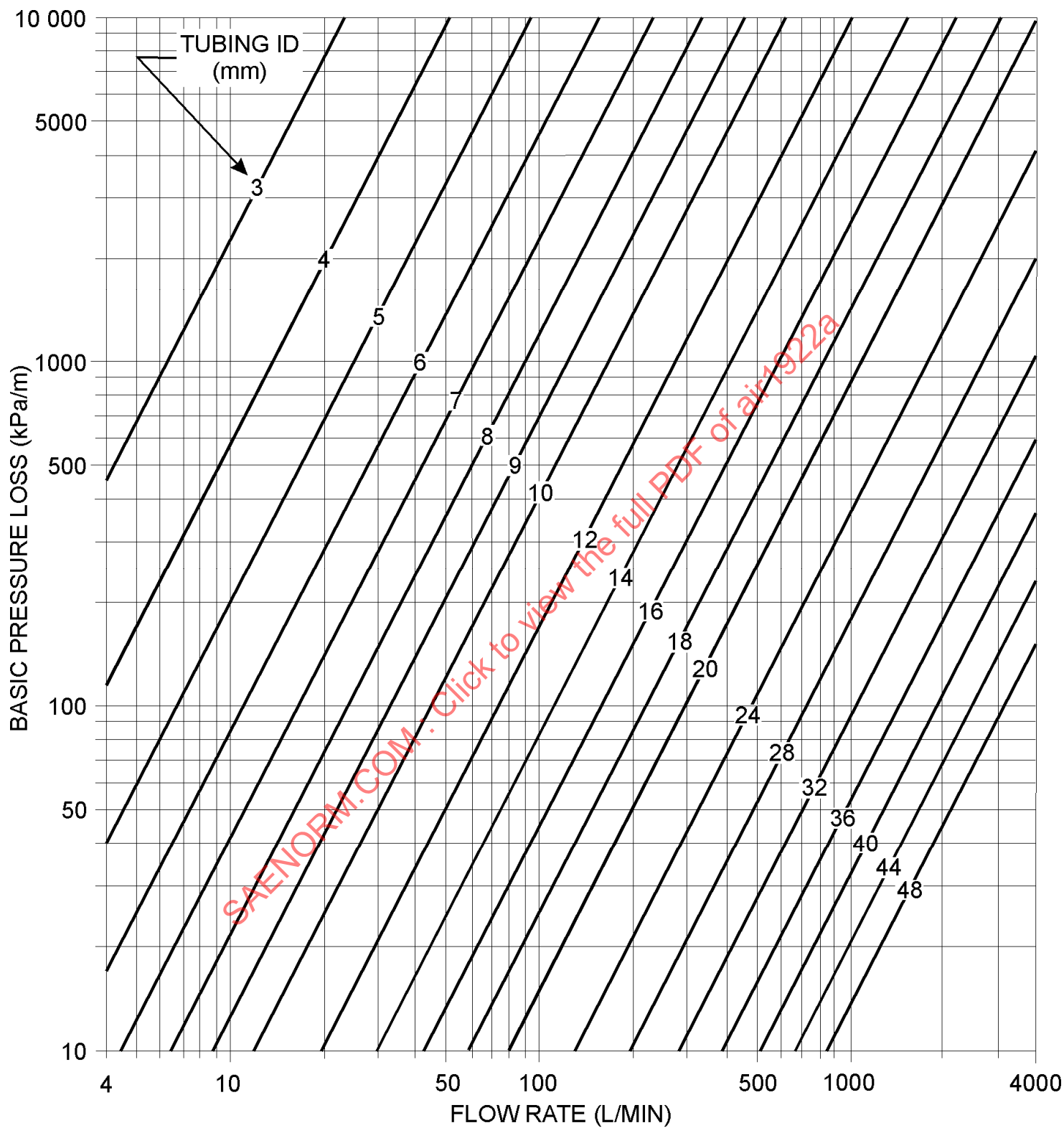


FIGURE 4B - PRESSURE LOSS IN HYDRAULIC LINES - TURBULENT FLOW - MIL-PRF-83282, T= 82 °C (180 °F)

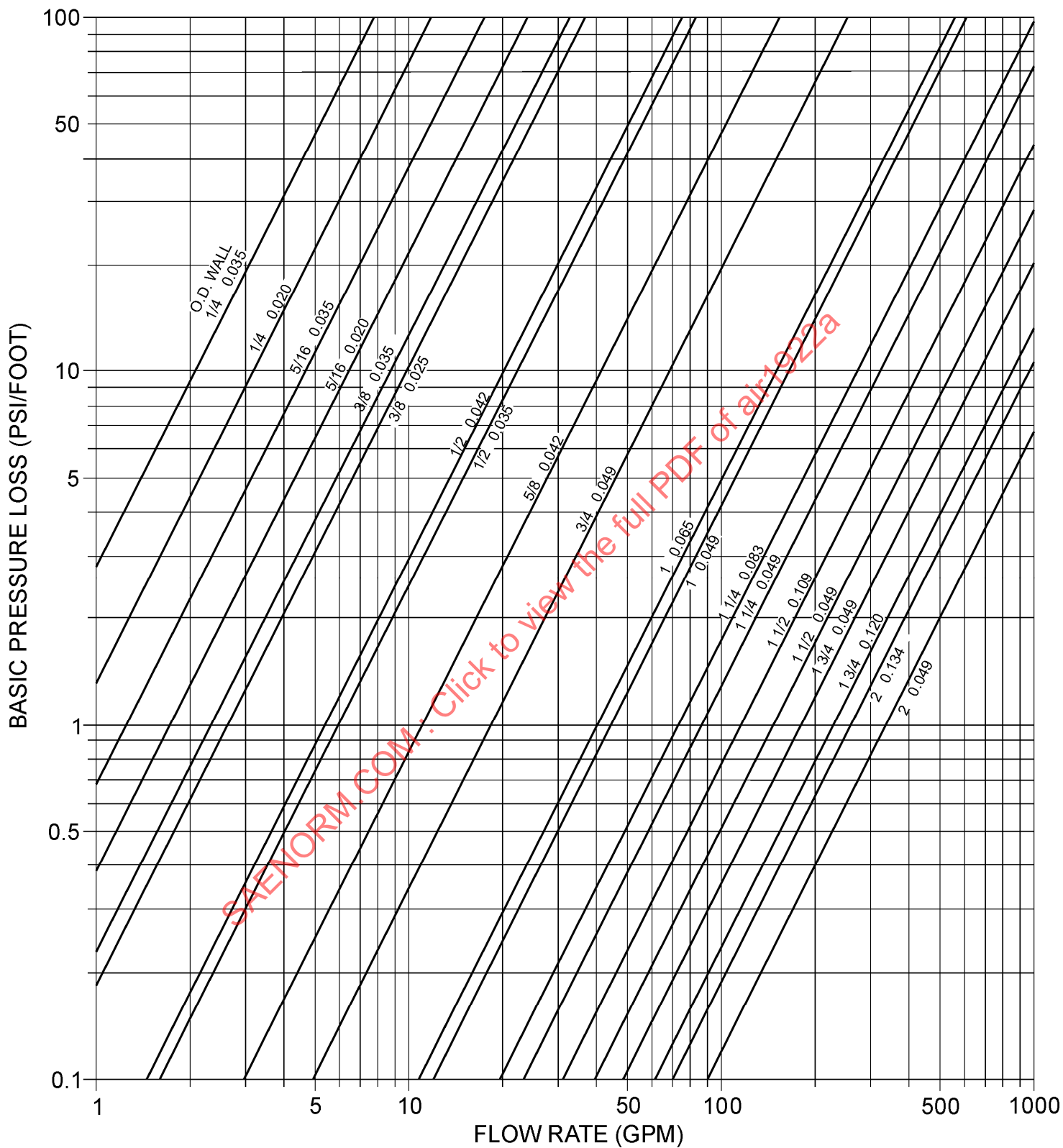


FIGURE 5A - PRESSURE LOSS IN HYDRAULIC LINES - TURBULENT FLOW - AS1241, T = 180 °F (82 °C)

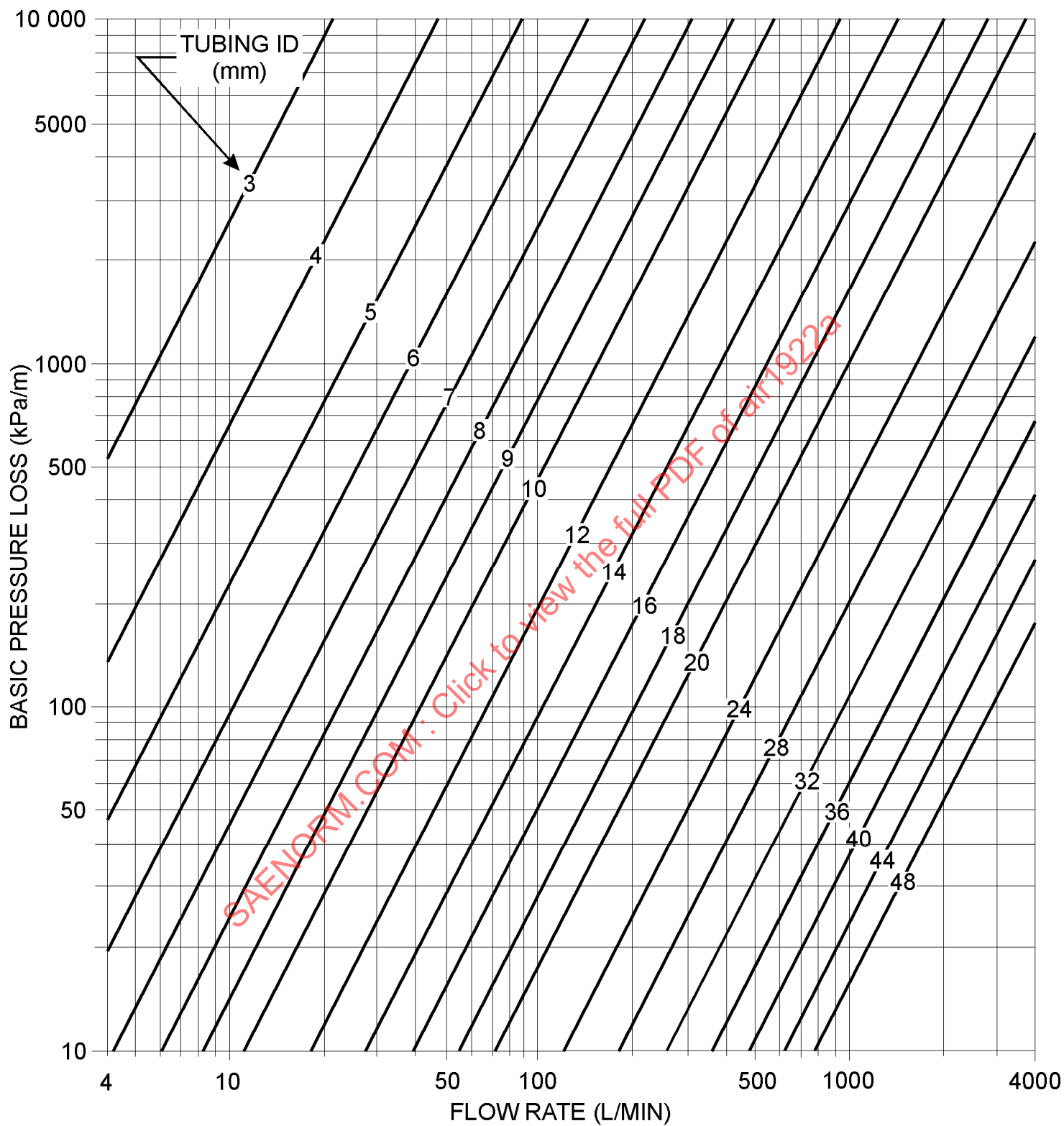


FIGURE 5B - PRESSURE LOSS IN HYDRAULIC LINES - TURBULENT FLOW – AS1241, T= 82 °C (180 °F)

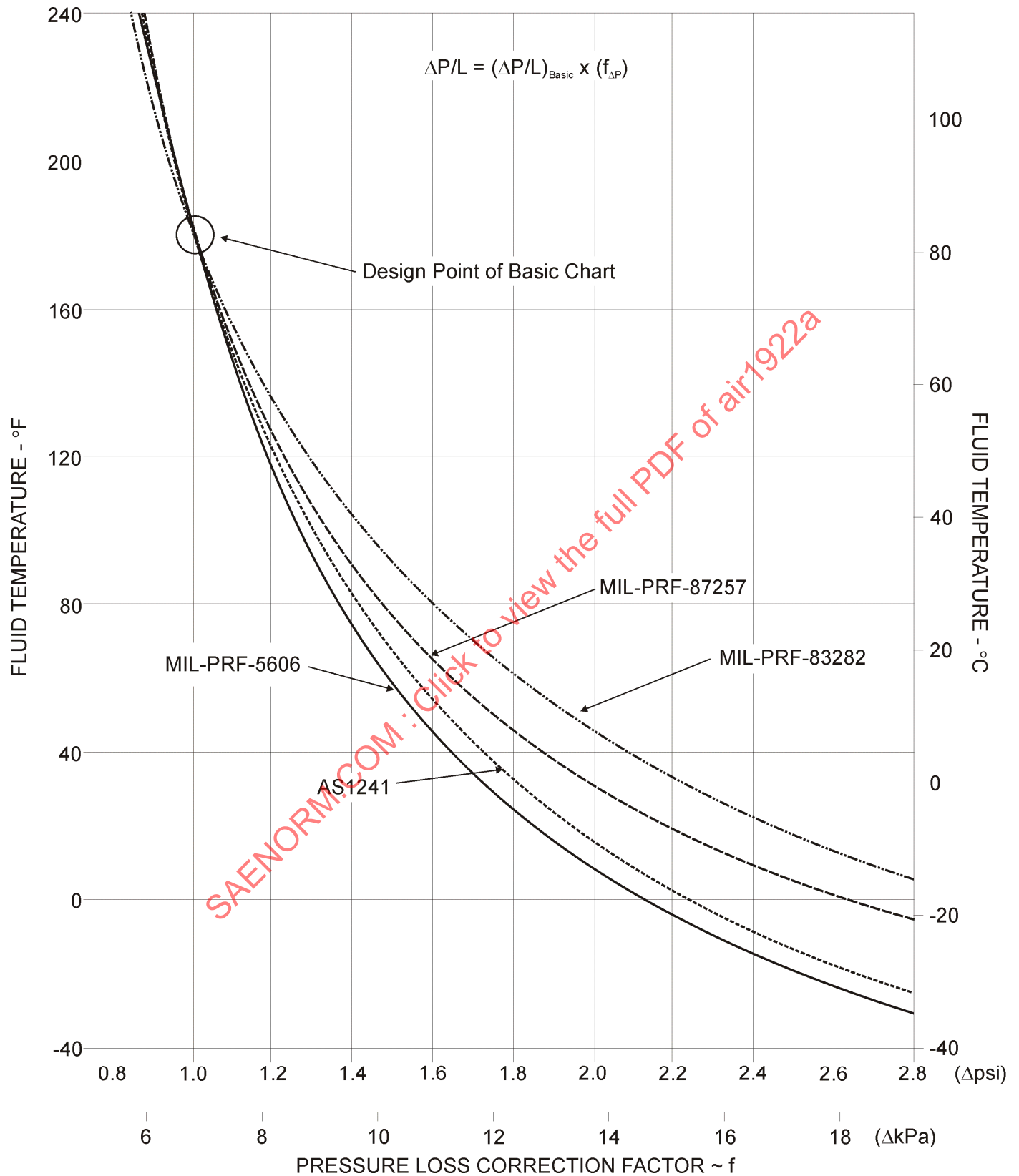


FIGURE 6 - EFFECT OF TEMPERATURE - TURBULENT FLOW

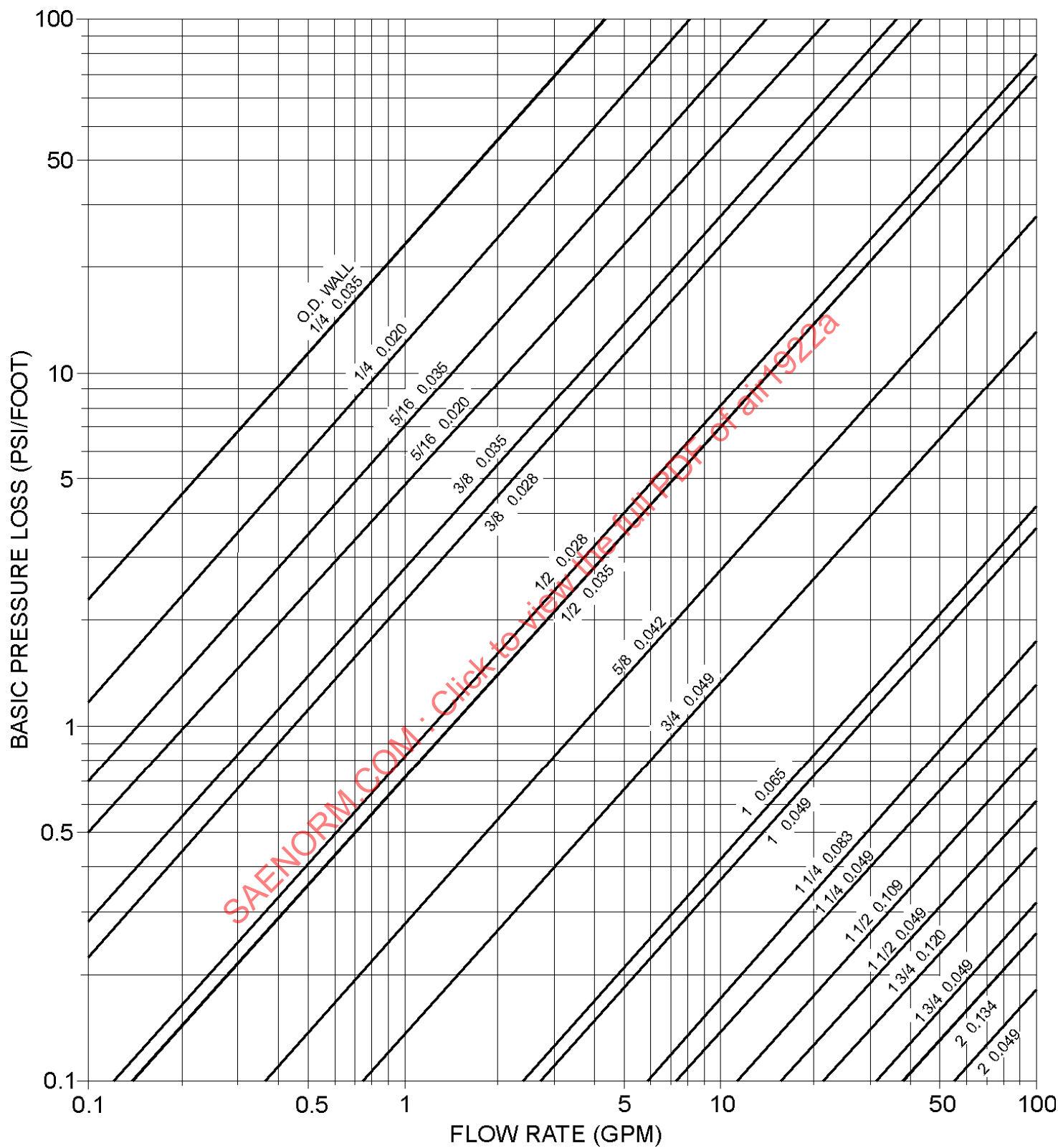


FIGURE 7A - PRESSURE LOSS IN HYDRAULIC LINES - LAMINAR FLOW - MIL-PRF-5606, T = 0 °F (-18 °C)

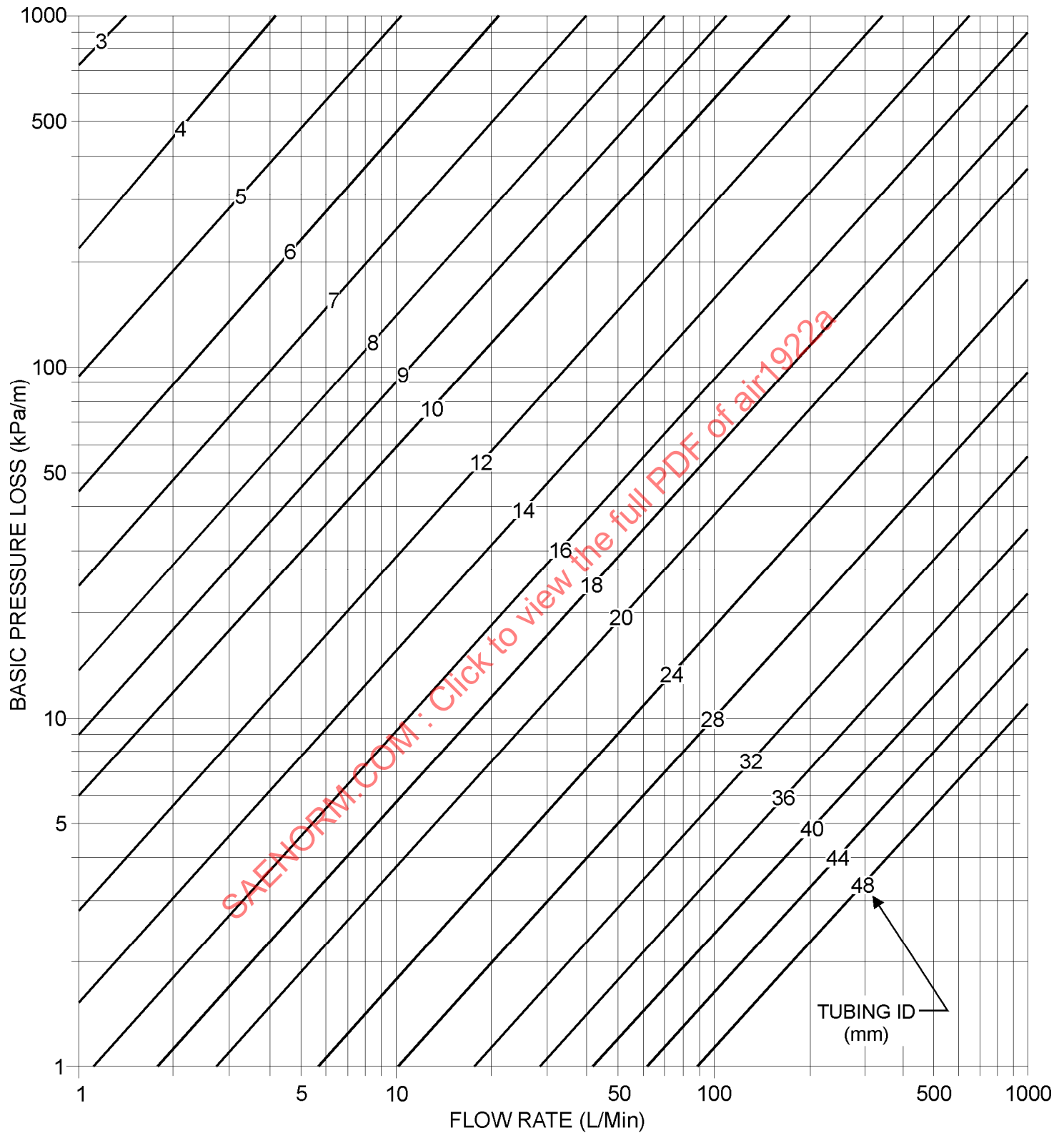


FIGURE 7B - FOR PRESSURE LOSS IN HYDRAULIC LINES - LAMINAR FLOW - MIL-PRF-5606, T = -18 °C (0 °F)

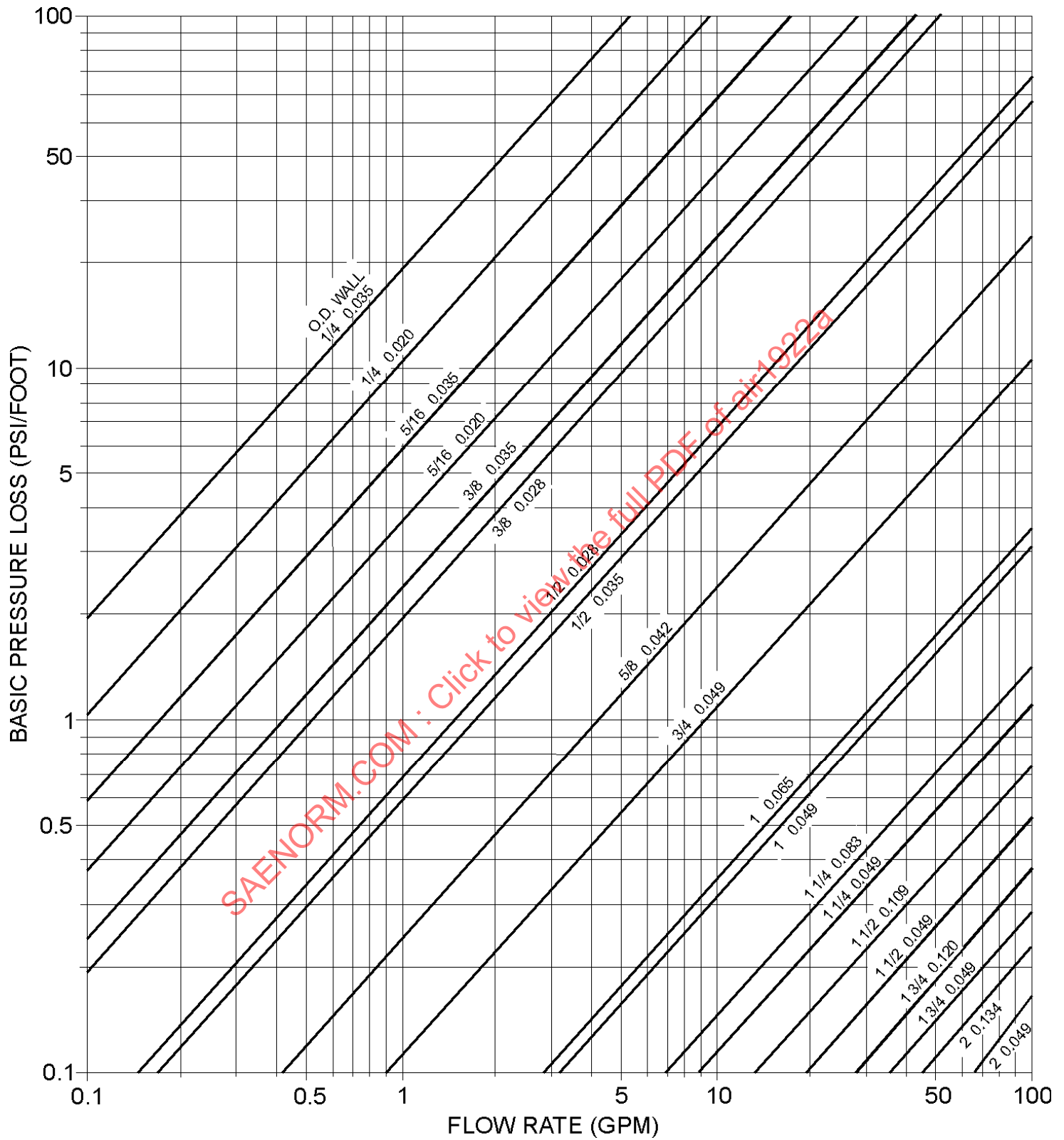


FIGURE 8A - PRESSURE LOSS IN HYDRAULIC LINES - LAMINAR FLOW - MIL-PRF-87257, T = 0 °F (-18 °C)

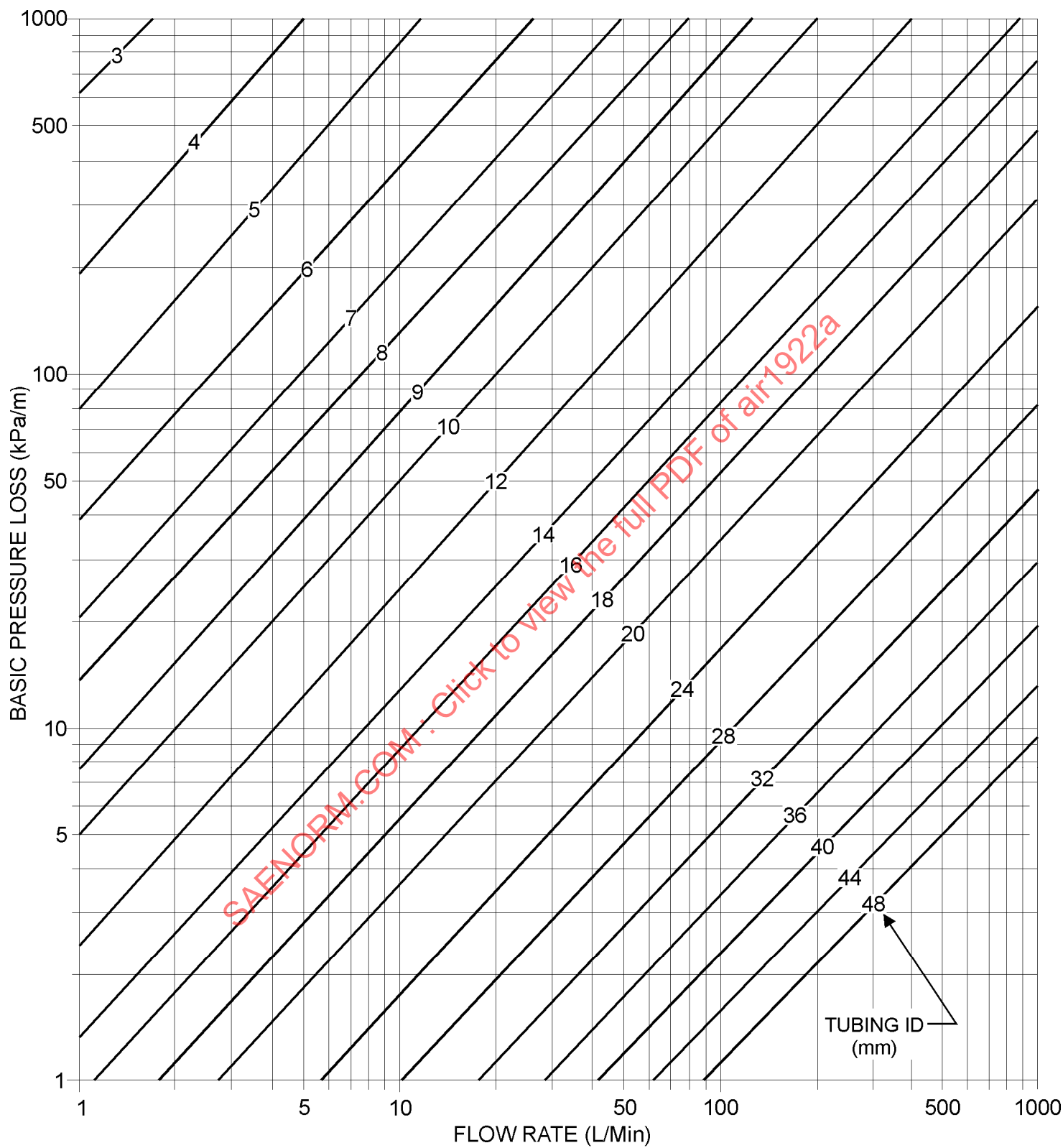


FIGURE 8B - PRESSURE LOSS IN HYDRAULIC LINES - LAMINAR FLOW - MIL-PRF-87257, T = -18 °C (0 °F)

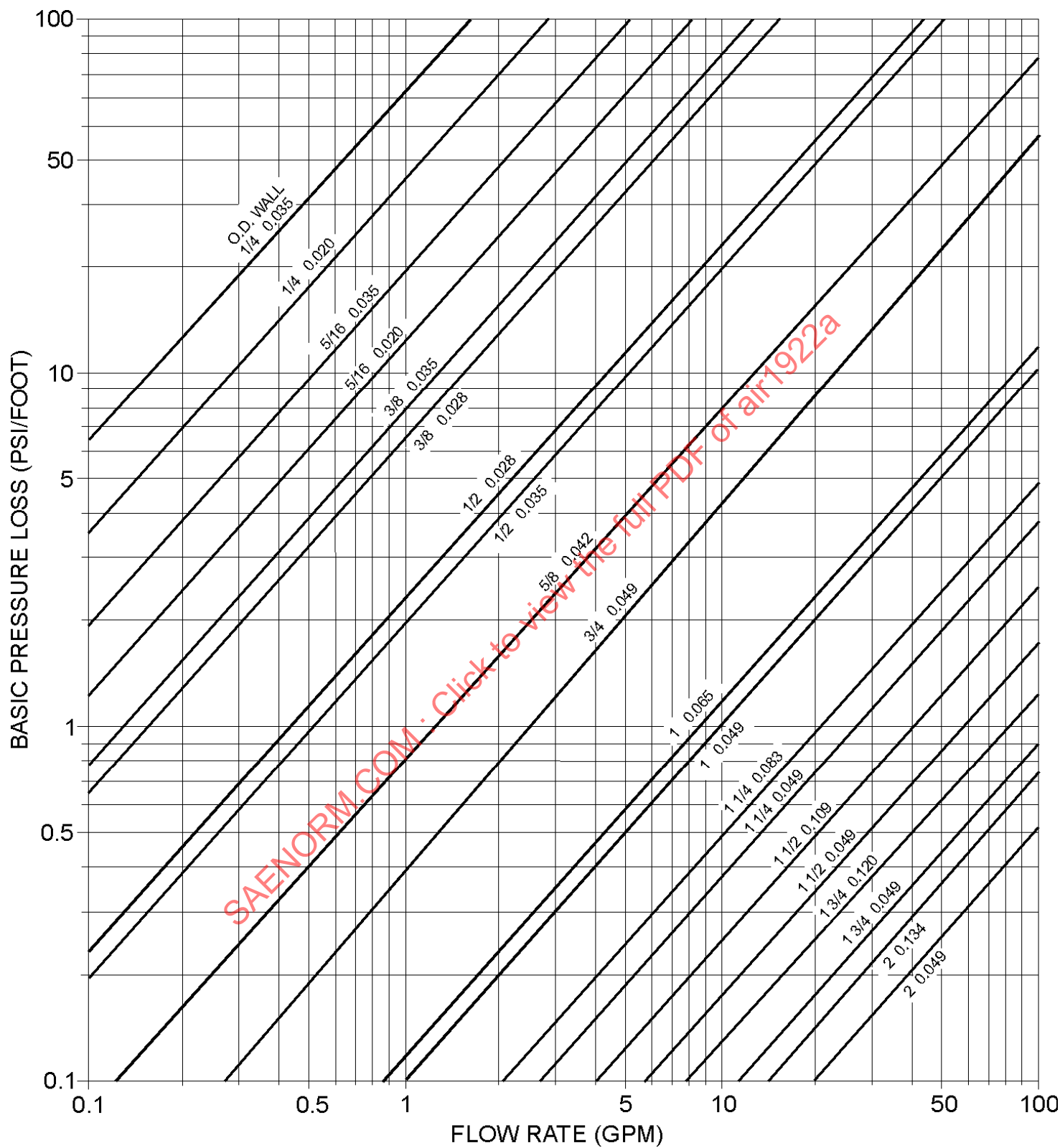


FIGURE 9A - PRESSURE LOSS IN HYDRAULIC LINES - LAMINAR FLOW - MIL-PRF-83282, T = 0 °F (-18 °C)

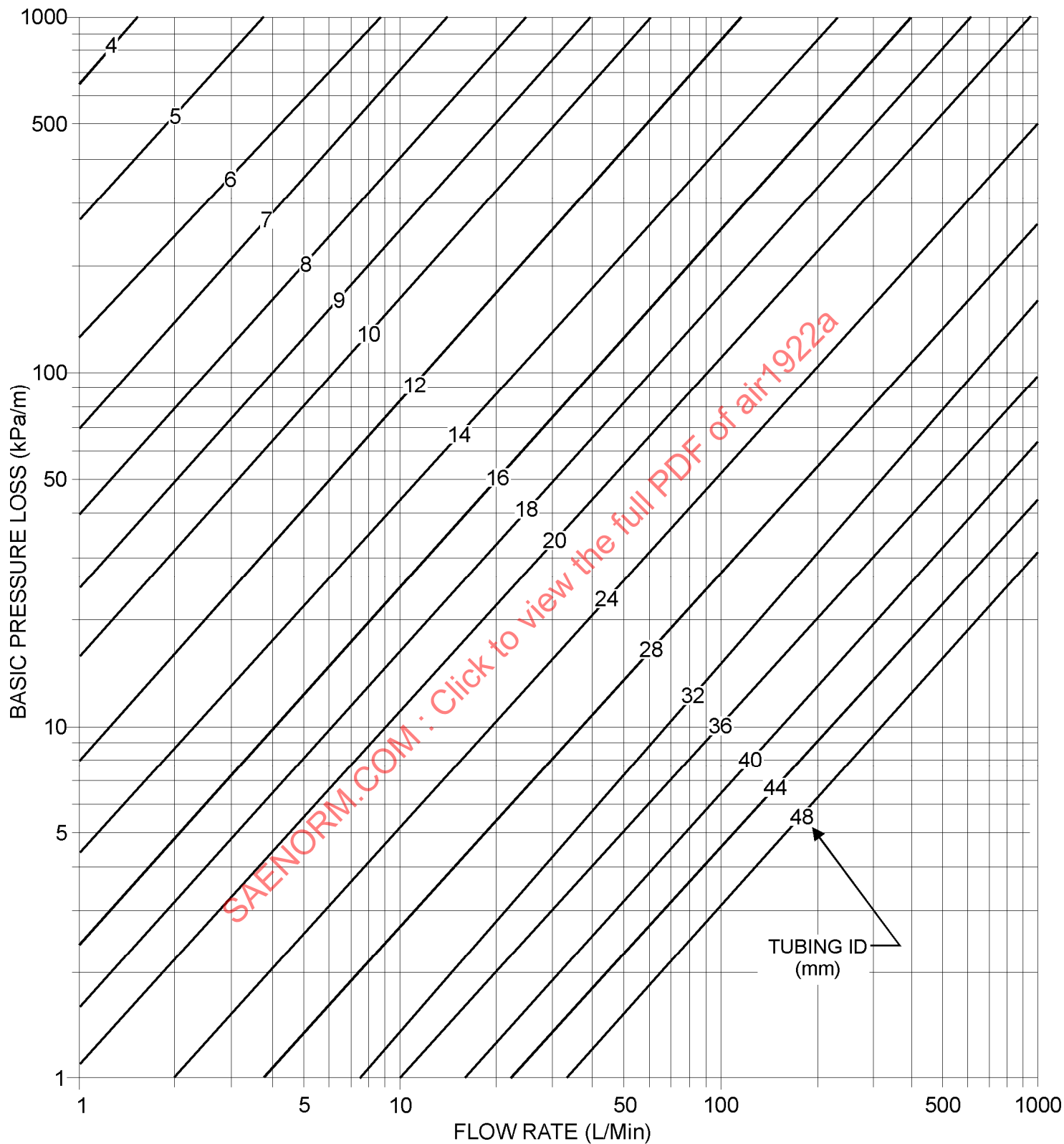


FIGURE 9B - PRESSURE LOSS IN HYDRAULIC LINES - LAMINAR FLOW - MIL-PRF-83282, T = -18 °C (0 °F)

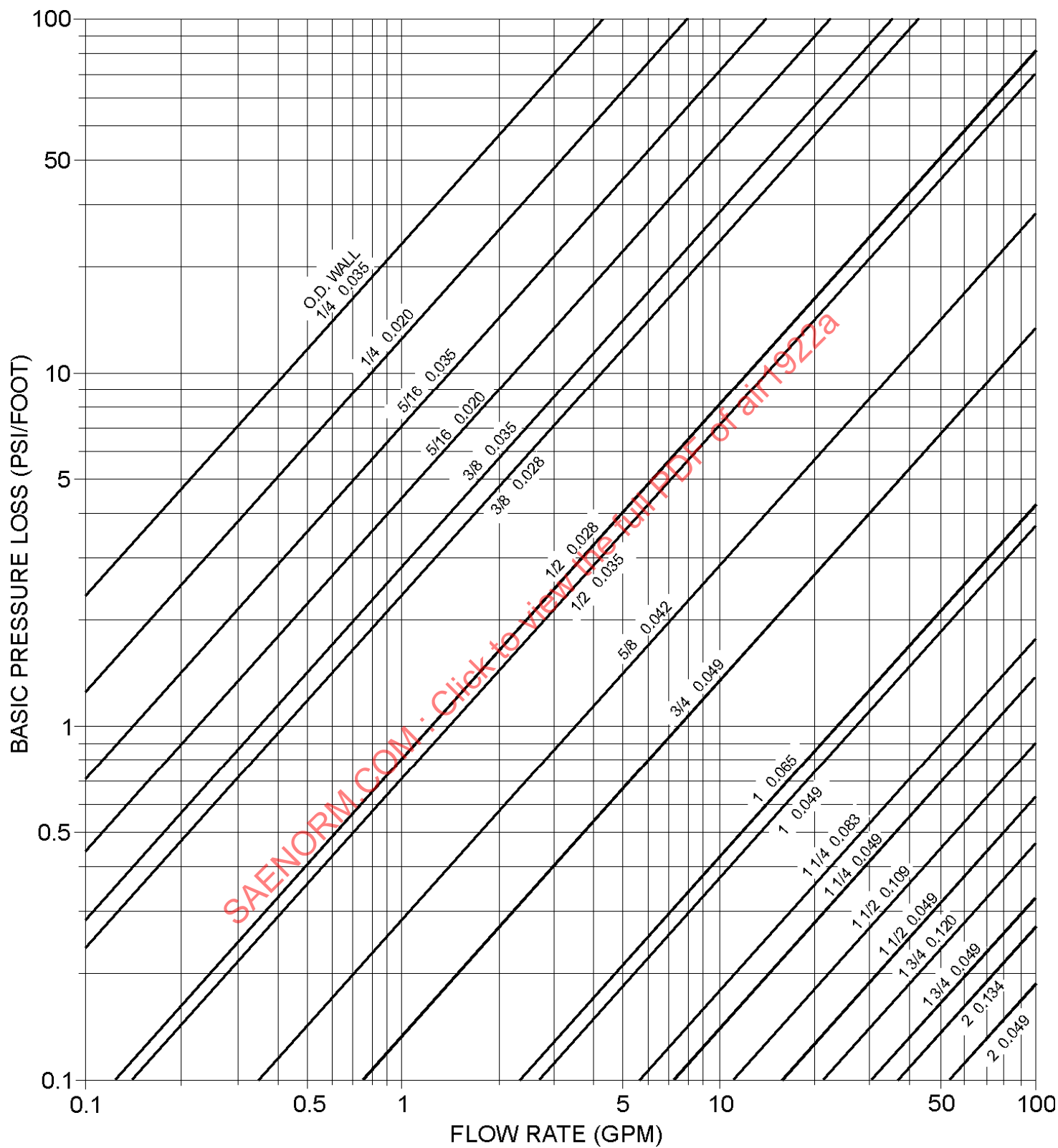


FIGURE 10A - PRESSURE LOSS IN HYDRAULIC LINES - LAMINAR FLOW – AS1241, T = 0 °F (-18 °C)

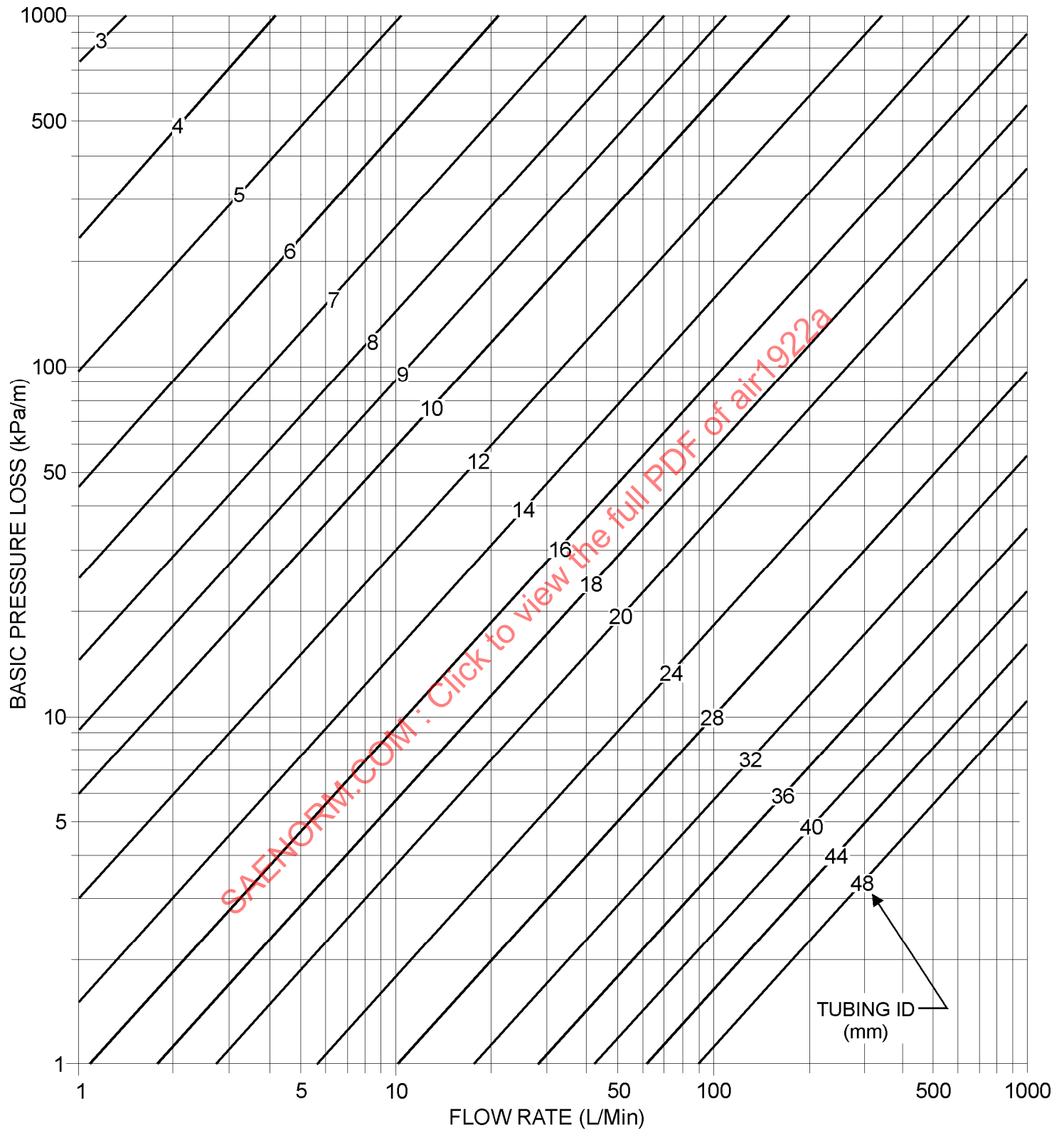


FIGURE 10B - PRESSURE LOSS IN HYDRAULIC LINES - LAMINAR FLOW – AS1241, T = -18 °C (0 °F)

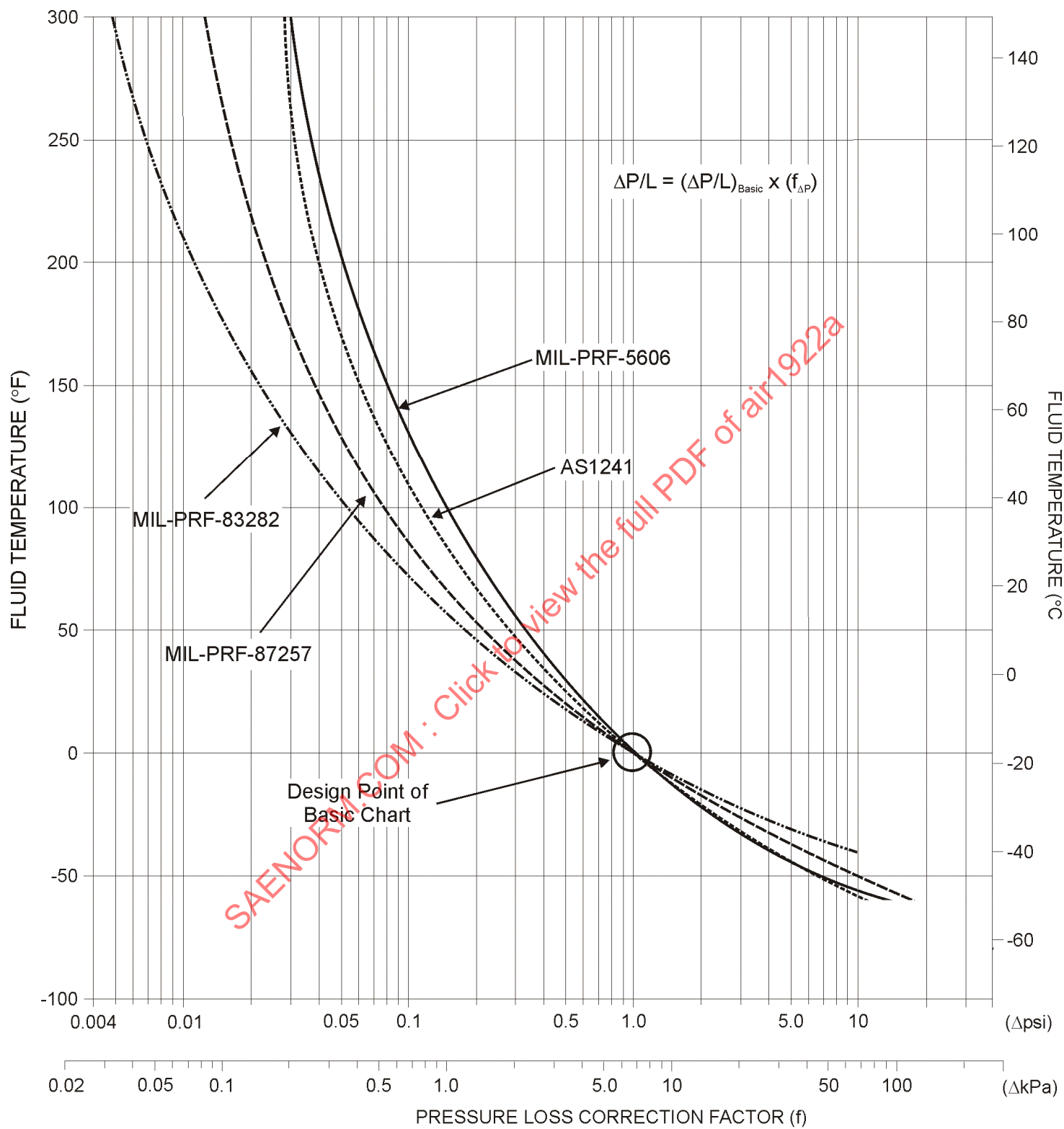


FIGURE 11 - EFFECT OF TEMPERATURE - LAMINAR FLOW

3.6.6 Example Calculation of Required Reservoir Pressure

- a. Hydraulic fluid, MIL-PRF-5606
- b. Operating temperature, -20 °F (-29 °C)
- c. Flow, 35 gpm (132 L/min)
- d. Inlet line size diameter OD, 1.25 in (32 mm)
- e. Wall thickness inlet line, 0.049 in (1.24 mm)
- f. Length inlet line, 18 ft (5.5 m)
- g. Effective inlet length, 20 ft (6.1 m)
- h. Critical inlet pressure, 35 psia (241 kPa abs)
- i. Response time, 0.050 s (50 ms)
- j. Pressure drop inlet components, 12 psi (83 kPa)
- k. Pump height over reservoir, 3 ft (.91 m)
- l. Fluid density, 55.7 lb m/ft³ (.89 kg/L) @ -20 °F (-29 °C); 52.2 lb m/ft³ (.84 kg/L) @ 120 °F (49 °C)

3.6.6.1 Find ΔP_{lin}

- a. From Figure 2a: the pressure loss at 180 °F (82 °C) = 0.2 psi/ft (4.5 kPa/m)
- b. From Figure 6:
 1. Temperature correction for -20 °F (-29 °C) = 2.5
 2. Net pressure loss = 0.5 psi/ft (11 kPa/m)
- c. From Figure 7a: the pressure loss at 0 °F (-18 °C) = 0.45 psi/ft (10 kPa/m)
- d. From Figure 11:
 1. Temperature correction for -20 °F (-29 °C) = 1.8
 2. Net pressure loss = 0.8 psi/ft (18 kPa/m)

Calculating the Reynolds number (refer to 3.6.7), flow becomes laminar, and 0.8 psi/ft (18 kPa/m) applies.

$$\Delta P_{lin} = 20 \text{ ft} \times .8 \text{ psi / ft} = 16 \text{ psi} \quad (\text{Eq. 11})$$

or

$$= 6.1 \text{ m} \times 20 \text{ kPa / m} = 122 \text{ kPa}$$

NOTES: 1. If the operating temperature were 120 °F (49 °C), the flow would remain turbulent, and the pressure drop would be 0.24 psi/ft (5.4 kPa/m).

2. The 20 ft (6.1 m) line length is effective length, and therefore already takes into account the effect of routing bends.

3.6.6.2 Find ΔP_{acc}

NOTE: For the purposes of this calculation, the non-uniformity of fluid acceleration has been ignored

a. English:

$$\begin{aligned}\Delta P_{acc} &= 88.1 \times 10^{-6} \rho \frac{L}{D^2} \times \frac{dQ}{dt} & (\text{Eq. 12}) \\ &= 88.1 \times 10^{-6} \frac{(55.7)(18)}{(1.152)^2} \times \frac{(35)}{(0.050)} \\ &= 46.6 \text{ psi (English)}\end{aligned}$$

b. Metric:

$$\begin{aligned}\Delta P_{acc} &= 21.2 \frac{\rho L}{d^2} \times \frac{dQ}{dt} & (\text{Eq. 13}) \\ &= 21.2 \frac{(0.89)(5.5)}{(29.3)^2} \times \frac{(132)}{(0.050)} \\ &= 319 \text{ kPa (metric)}\end{aligned}$$

3.6.6.3 Find ΔP_{sta}

$$\Delta P_{sta} = 0.007 \rho h_1 = 0.007 \times 55.7 \times 3 = 1.2 \text{ psi (English)} \quad (\text{Eq. 14})$$

or

$$= 9.8 \rho h_1 = 9.8 \times .89 \times .91 = 7.9 \text{ kPa (metric)}$$

3.6.6.4 Find P_{resv}

$$P_{resv} = P_{crit} + \Delta P_{acc} + \Delta P_{com} + \Delta P_{lin} + \Delta P_{sta} \quad (\text{Eq. 15})$$

$$= 35 + 46.6 + 12 + 16 + 1.2 = 111 \text{ psia (English)}$$

or

$$= 241 + 319 + 83 + 110 + 7.9 = 761 \text{ kPa (metric)}$$

3.6.6.5 Alternative Assumptions/Requirements

Even though the following calculations make some approximations, the results are of practical use and highlight the significant effects of:

- Fluid acceleration
- Fluid temperature
- Just one line size change

3.6.6.5.1 Non-Uniformity of Fluid Acceleration

This affects the ΔP_{acc} term

a. English:

$$\begin{aligned}\Delta P_{acc} &= 176.2 \times 10^{-6} \times \rho \times \frac{L}{D^2} \times \frac{dQ}{dt} & (\text{Eq. 16}) \\ &= 176.2 \times 10^{-6} \times \frac{(55.7)(18)}{(1.152)^2} \times \frac{(35)}{(0.050)} \\ &= 93.2 \text{ psi (English)}\end{aligned}$$

$$P_{resv} = P_{crit} + \Delta P_{acc} + \Delta P_{com} + \Delta P_{lin} + \Delta P_{sta} \quad (\text{Eq. 17})$$

$$P_{resv} = 35 + 93.2 + 12 + 16 + 1.2 = 157 \text{ psia (English)}$$

b. Metric:

$$\begin{aligned}\Delta P_{acc} &= 42.4 \times \frac{\rho L}{d^2} \times \frac{dQ}{dt} \\ &= 42.4 \times \frac{(0.89)(5.5)}{(29.3)^2} \times \frac{(132)}{(0.050)} \\ &= 638 \text{ kPa (metric)}\end{aligned}$$

$$P_{resv} = P_{crit} + \Delta P_{acc} + \Delta P_{com} + \Delta P_{lin} + \Delta P_{sta}$$

$$= 241 + 638 + 83 + 110 + 7.9 = 1080 \text{ kPa (metric)}$$

3.6.6.5.2 Higher Fluid Temperature

Had full operation been a requirement at say, 120 °F (49 °C) or above, the following values would have applied:

- a. $P_{crit} = 30 \text{ psi (207 kPa)}$
- b. $\Delta P_{acc} = 43.7 \text{ psi (301 kPa)}$
- c. $\Delta P_{com} = 9 \text{ psi (62 kPa)}$
- d. $\Delta P_{lin} = 5 \text{ psi (34 kPa)}$
- e. $\Delta P_{sta} = 1.1 \text{ psi (8 kPa)}$

$$\begin{aligned}
 P_{resv} &= P_{crit} + \Delta P_{acc} + \Delta P_{com} + \Delta P_{lin} + \Delta P_{sta} \\
 &= 30 + 43.7 + 9 + 5 + 1.1 = 89 \text{ psia (English)} \\
 &= 207 + 301 + 62 + 34 + 8 = 612 \text{ kPa (metric)}
 \end{aligned}$$

3.6.6.5.3 Increased Inlet Tube Diameter

Had an inlet line of 1.50 in (38 mm) diameter been used and with the commensurate inlet components, the operation at 120 °F (49 °C) would have required approximately 72 psia (496 kPa abs.) at the reservoir for the above example.

3.6.7 Hydraulic Line Losses

For most airborne hydraulic power transmission systems, the flow rates and line sizes will be such that turbulent flow will exist during normal operation. During off-design operation at very low power, laminar flow might exist; but the pressure drop for such a case would be trivial. In the case of hydraulic starting systems for jet engines, however, the entire duty cycle may be completed before the system temperature is raised appreciably above its original level. For such systems, laminar flow may be present at large flow rates, and must be considered in the system analysis.

For laminar and turbulent pipe flow, the following expressions may be used:

a. Reynolds Number

$$R = \frac{VD}{\nu'} = 3163 \frac{Q}{\nu D} \text{ (English)} \quad (\text{Eq. 18})$$

$$R = 21220 \frac{Q}{\nu D} \text{ (metric)}$$

b. Friction Factor (English and metric)

1. Laminar flow

$$f_{lam} = 64/R \quad (\text{Eq. 19})$$

2. Turbulent flow

$$f_{turb} = .316/R^{.25} \quad (\text{Eq. 20})$$

c. Pressure Drop Per Line Length

$$\frac{\Delta P}{L} = 2.16 \times 10^{-4} f \rho \frac{Q^2}{D^5} \text{ (English)} \quad (\text{Eq. 21})$$

$$\frac{\Delta P}{L} = 2.25 \times 10^{-5} f \rho \frac{Q^2}{D^5} \text{ (Metric)}$$

where:

Q = Volume flow, gpm (L/min)

D = Line inside diameter, in (mm)

ν' = Fluid viscosity (kinematic), in²/s (mm²/s)

ν = Fluid viscosity (kinematic), centistokes (cs)

ρ = Fluid density, lbm/ft³ (kg/L)

ΔP = Pressure drop, lbf/in² (kPa)

L = Length, ft (m)

V = Flow velocity, in/s (m/s)

F = Friction factor

NOTE: R < 2100 is considered laminar, R between 2100 and 4000 is considered to be the transition between laminar and turbulent, and R > 4000 is considered turbulent.

When determining system pressure drop, the appropriate flow condition may be ascertained by finding the pressure drop for both laminar and turbulent flow at the given temperature, line size, and flow rate, then using the larger pressure drop value.

4. PUMP OUTLET LINE

Factors in the pump outlet (pressure) line that affect pump life include:

- a. Pressure pulsations
- b. Pump-system response
- c. Using a check valve in the pump outlet line
- d. Parallel pump installation

4.1 Pressure Pulsations

4.1.1 General

The pulsations in the output flow of a piston-type hydraulic pump are transmitted throughout the pressure side of the hydraulic system. The dynamic resistance (impedance) of the hydraulic system fluid translates the flow pulsations into pressure pulsations, also known as pressure ripple.

Unsteady flow is produced by a positive-displacement pump due to two factors:

- a. The kinematic flow ripple arises due to the summing of the sinusoidal output flows of the pumping elements. For pumps with nine or more pumping elements, the kinematic flow ripple is approximately 1.5%. The frequency of the kinematic flow ripple for pumps with an odd number of pistons is 2 times the piston pass frequency.

The piston pass frequency is equal to the product of the drive speed and the pumping elements.

Example: A nine piston pump rotating at 5000 rpm has a piston pass frequency of 750 Hz.

$$\frac{9 \text{ pistons} \times 5000 \text{ rpm}}{60 \text{ s / min}} = 750 \text{ Hz} \quad (\text{Eq. 22})$$

- b. The other much larger source of flow ripple is due to the compressibility of the fluid. The opening of the pumping element that is transitioning from low pressure to high is delayed in order to raise the chamber pressure to near the system pressure. This is called the pump timing. This delay, together with any mismatch in system pressure and pump chamber pressure when the pumping element is open, causes flow oscillation.

In addition to the piston pass frequency as determined above, significant pulsation amplitudes are often observed at the first, second and third harmonics as well, and these should also be considered in the design of the system installation. Hydraulic flow/pressure pulsations excite high frequency mechanical motion in lines and components. When the frequency (speed) and harmonics of the acoustic source (pump) is equal to a natural frequency of the hydraulic fluid column, a resonance condition occurs. Pressure pulsations at resonance may be very high, for example, ± 1000 psi (6900 kPa) in a 3000 psi (20 700 kPa) system. These resonant conditions are potentially very destructive to the system hardware as excessive motion causes stresses that may result in failure of lines, components and/or mounting hardware. In extreme cases, the pressure vessel (lines or components) may reach fatigue failure limits in a few minutes. Therefore changes to the system and/or pump may be required to reduce pressure pulsations at system resonant frequencies when they occur at primary operating conditions (e.g., idle, cruise, etc.) but not necessarily at transient conditions.

Consideration must be given to operation outside the nominal operating conditions of speed, pressure or temperature ranges. The pump and system may be compatible under the normal operating conditions but affect resonances under other conditions that produce high levels of pressure ripple. These may be of short duration during the start-up phase that it is of no serious concern but any quasi-persistent condition should be treated with care, as the pulsation frequencies involved are high. A high number of cycles can be accumulated over the life of an aircraft.

Pressure pulsations in the outlet line can also have an effect on pump life as high frequency resonant pulses reflected back to the pump can cause the pump compensator mechanism to oscillate at the high frequency, with resultant rapid wear of the pump parts. Test data has illustrated that lower pressure pulsations result in lower pulses being reflected back to the pump from the system. It has also been demonstrated in some systems that higher pressure pulsations reduce pump life as well as the life of other components. The failure of components downstream of the pump can also result in reduced pump life.

Pressure pulsations also occur in the pump inlet/return system. Normally, the inlet side of the system is not a problem because of the low pressure and low energy level at which it operates. However, thin wall tubing, high installation stresses, and an adverse resonance condition could produce failures in the inlet side of the system.

The coupling of hydraulic to mechanical resonances is a complex phenomenon, and is a function of line size and material, configuration (routing), and installation constraints. The total stresses in lines are a combination of hoop stress from internal pressure and bending stress due to installation and induced vibration. Large pump outlet and inlet lines having an outside diameter (OD) of 1 in (25 mm) and above are particularly vulnerable to high installation stress. The total combined line stress must be low enough to provide infinite fatigue life.

4.1.2 Acceptable Level of Pressure Pulsations

An acceptable level of pressure pulsation in one system may not be acceptable in another system due to differences in mechanical response. High frequency line motion induced by pressure pulsations cannot be controlled by normal line support techniques such as clamps. Clamps must be designed to withstand the line vibration without wearing out the cushion between the clamp and line or causing chafing through the line. Pressure pulsation requirements are often specified to be <10% and <5% (peak-to-peak) of rated pressure for military and commercial aircraft respectively.

The following relates central system pulsation level to potential problems.

- a. >1100 psi (>7586 kPa) (peak-to-peak): Rapid failure of pump discharge line due to pressure and vibration stresses. Possible failures of mounting structure and internal functions of central components
- b. >600 psi (>4138 kPa) (peak-to-peak): Line clamp cushion wear out, line failure due to clamp chafing, poor clamp life, frequent inspections required, discharge line check valve wear out

The first category (>1100 psi (>7586 kPa) (peak-to-peak)) is a potential safety of flight situation. The second category (>600 psi (>4139 kPa) (peak-to-peak)) is one of nuisance level problems that probably surfaces only after a considerable amount of operational flight experience. The best approach is to verify the acceptable line stresses on the systems laboratory simulator and the first flight aircraft to preclude failures of the first category. Changing the line length from the pump outlet port to the filter or other simple plumbing changes may be identified to relocate resonances away from continuous operating speeds.

If stress levels due to pressure pulsations are not acceptable, wide-band attenuators or hoses for mechanical decoupling must be considered. Hoses may provide mechanical decoupling which can eliminate clamp/line wear out problems. However, one must accept the penalties when using hoses instead of tubing, such as limited life, added weight and cost, and larger diameters.

It should be noted that away from the high acoustic energy of the pumps, gearboxes, and engines, the vibration environments are relatively benign. Good line support and adequate clearances between lines, and lines and structure, precludes significant vibration related problems.

4.1.3 Means to Reduce Pressure Pulsations

Techniques that significantly reduce system pressure pulsations over the operating speed range of the pump include attenuation components such as:

- Adding a flow-through volume in the pump outlet port - this can generate anti-phase pulsations that will damp out most of the pulsations generated by the pump
- Adding a stand-off volume (such as a teed-off line) downstream of the pump outlet port - this can also generate anti-phase pulsations, but at only at a specific pump speed

NOTE: The attenuating means should be located as close to the source (pump) as possible for maximum protection of the downstream system as well the pump itself. If attenuators are to be installed at some downstream distance from the pump, the hydraulic line between the pump and attenuator must be checked to verify that no resonant conditions exist.

- Adding a filter close to the outlet port - the filter element absorbs all the pressure pulsations and provides a ripple free output

- Adding an accumulator to the central system which can damp out any resonances in the pressure line
- Making some detail changes inside the pump to fine tune the pump to the system

Restrictors may also be used to attenuate pulsations in a dead-ended line such as that terminated by a static pressure transmitter. When a transmitter must be located at the end of a line, a small restrictor should be placed at the acoustic source (i.e., the "T" where the dead-end line begins). This may protect the entire line and transmitter from significant acoustical energy. A secondary benefit is attenuation of system transient pressure spikes that may cause erroneous cockpit indications on a poorly damped pressure gage.

A check valve may be used to "acoustically" isolate a dead-end ground service line that leads to the system pressure supply. Again, the check valve must be placed at the downstream end of the service line adjacent to the main line. This will isolate the service line from pressure pulsations generated by the onboard system pump during normal aircraft operation.

4.2 Hydraulic Pump Response Characteristics

Each variable displacement hydraulic pump is a highly under-damped servomechanism with a transient response that is dependent on many parameters. System and pump characteristics that affect response (time rate of pressure change (dp/dt)) are:

- a. Fluid bulk modulus
- b. Fluid volume under compression
- c. Line characteristic impedance
- d. System load characteristics
- e. Pump speed
- f. Pump leakage
- g. Pump flow gain
- h. Pump/compensator gain
- i. Initial steady state pressure and flow

These factors also play a critical part in determining pump stability. When system flow demands change, corresponding transient pressure changes will occur. These perturbations in line pressure are sensed by the pump pressure compensator that then tries to compensate for the changes in pressure by adjusting the flow.

The change in flow depends on how far the line pressure is away from a reference value set in the pump. The speed of response is defined as the time required for the pump control to maintain pressure in either direction (zero flow to full flow or full flow to zero flow demand) in the pressure compensated range.

Because of the many parameters that affect pump response and stability, the hydraulic system designer must adequately specify the circuit and the performance required. The pump performance should be such that any pressure overshoot is limited when pump demand is reduced from full flow to zero flow. Figure 12 presents a typical transient pressure – time characteristic as the pump responds to the system flow demand from full flow to zero flow. Under the same conditions, the system relief valve is to limit the overshoot should the pump compensator fail. On sudden demands of flow, such as zero flow to full flow, the pump should respond to limit system pressure reduction to a minimum value. Figure 13 presents a typical transient pressure – time characteristic as the pump responds to the system flow demand from zero flow to full flow. The pump should also provide 90% of steady state full flow pressure within a specific time period. Note that the design of the inlet line also plays a key role in this characteristic.

More information on pump response characteristics is contained in AS595 and AS19692.

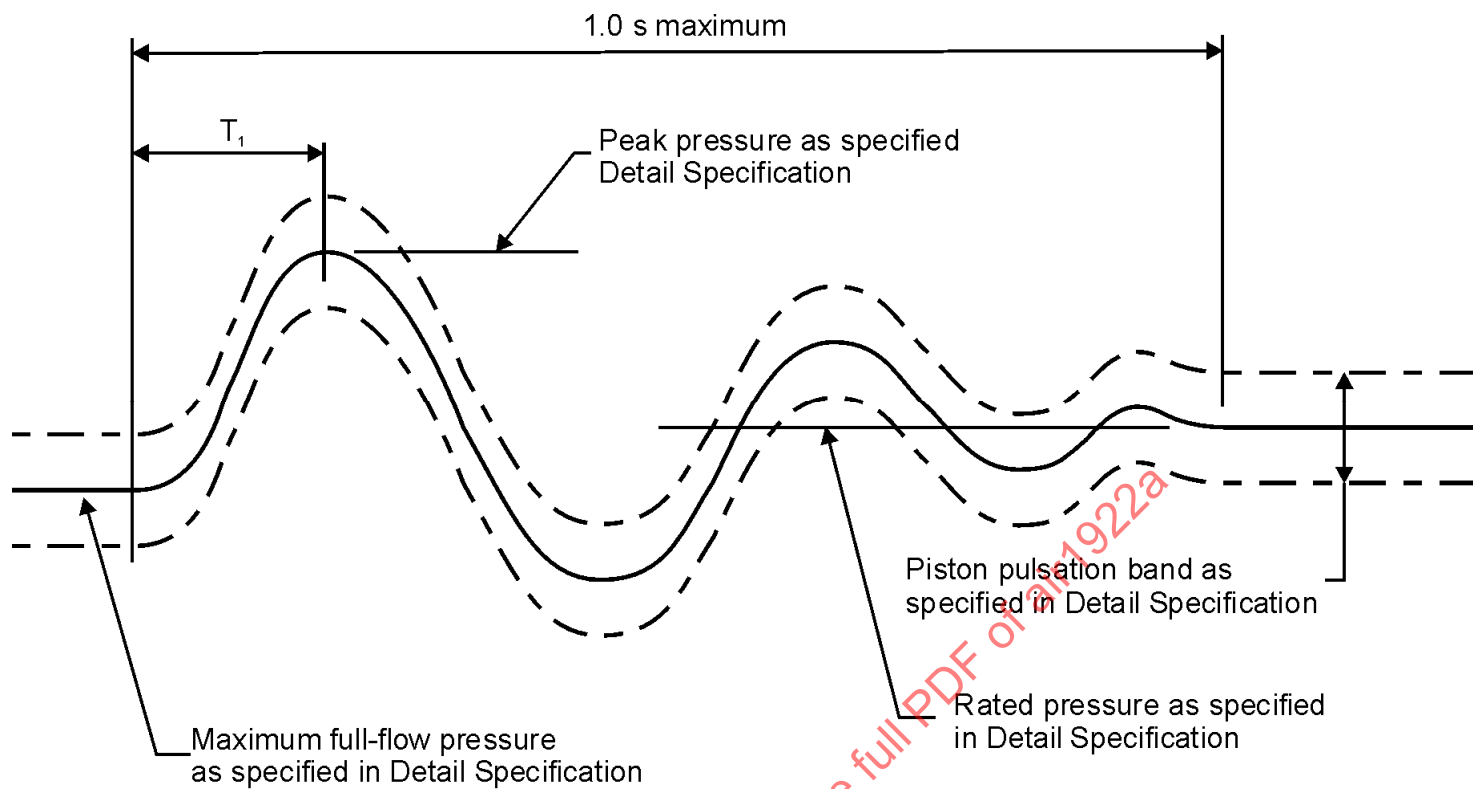


FIGURE 12 - TYPICAL PRESSURE TRANSIENT FULL FLOW TO ZERO FLOW

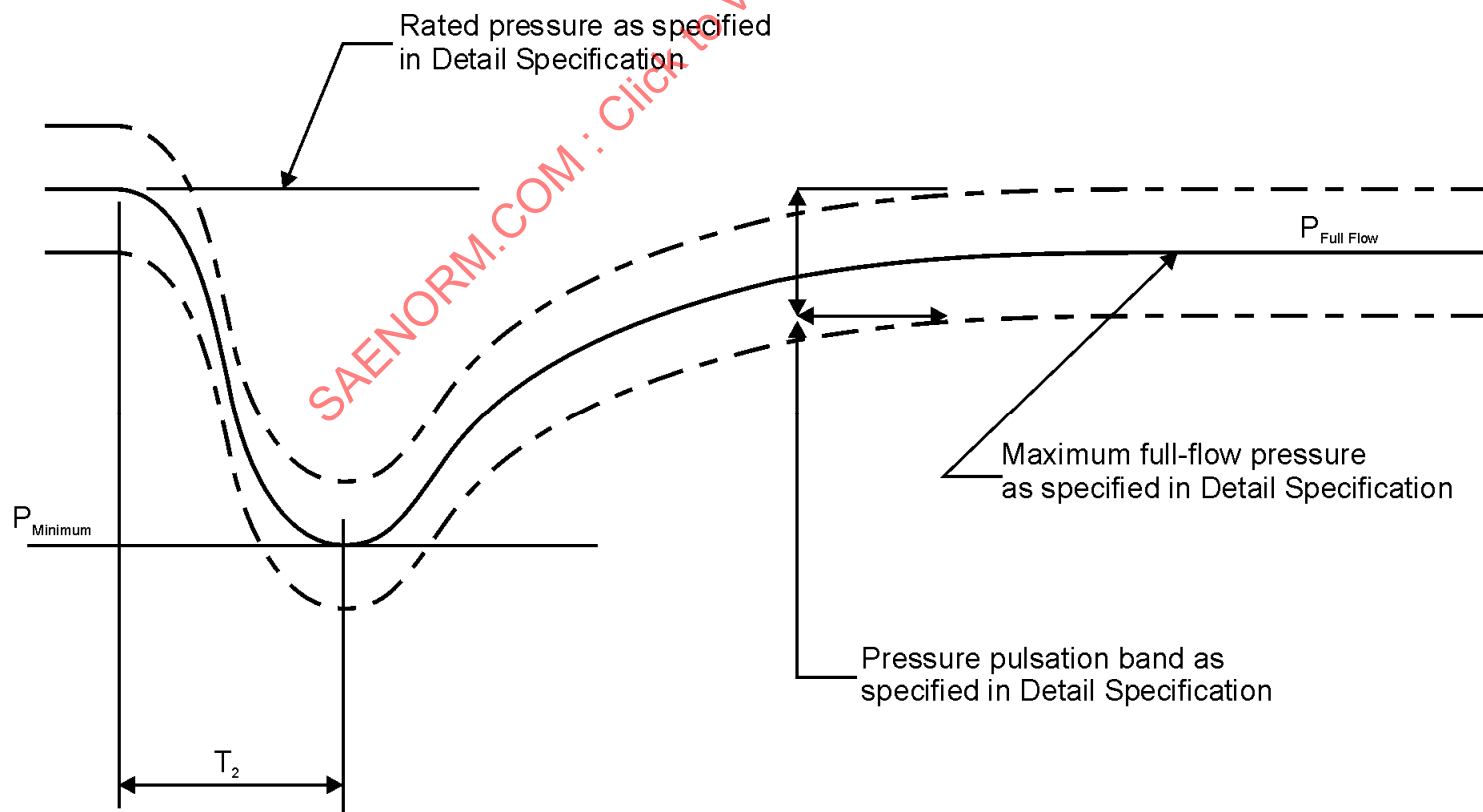


FIGURE 13 - TYPICAL PRESSURE TRANSIENT ZERO FLOW TO FULL FLOW

4.3 Check Valve in Pump Outlet Line

The use of a check valve in the pump outlet line is to prevent back flow that can cause back drive motoring (such as when external hydraulic power supply ground support equipment is pressurizing the hydraulic system while the engine-driven pump is not rotating) of the pump and protect the pump from adverse system pressure spikes.

The location of a check valve in the outlet line can greatly affect the resonant characteristics of the pump and the system. This is particularly significant in systems that have zero flow conditions. During the transition from a flow to a zero flow condition the high transient pressure produced by the pump passes over the check valve. When the pump regulates its output pressure back to nominal the check valve prevents back flow from the system, trapping this higher transient pressure in the system, which has two effects:

- a. The system pressure indication is too high
- b. The pump is exposed to a very small system volume up to the check valve which may create stability problems

Even without the transient pressure effect, a check valve that is located very close to the pump may integrate normal pressure ripple of the pump until the pressure recorded downstream of the check valve reaches the level of the maximum peak of the pressure ripple.

4.4 Parallel Pump Installations

In parallel pump installations, means should be provided to have a master/slave relationship between the two pumps in order to prevent any interactions with each other, and hence eliminate any problems of pump instability.

One way to operate two parallel pumps in this master/slave relationship is to install a higher cracking pressure check valve in the outlet line of one of the pumps. For small flow demands, the pump with the lower cracking pressure check valve will supply the flow; however when a large flow demand occurs, the system pressure will reduce which then enables the pump with the higher cracking pressure check valve to also supply fluid.

The check valve should be designed with adequate damping to preclude poppet instability around the cracking point.

5. PUMP CASE DRAIN LINE

5.1 General

The function of the pump case drain line is primarily the removal of internal leakage from the pump. In so doing, the case drain line provides a means of rejecting heat from the unit, and offers a means of monitoring pump performance. Factors to be considered in the design of the case drain lines include the normal hydraulic tubing practices that are observed for all hydraulic system lines in addition to the guidelines and factors which follow.

NOTE: Some pumps may not have an external case drain line. Rather, the pump case is connected internally to the low pressure suction port. This configuration unit may only be used in applications in which the pump is always delivering some output flow to the system high pressure line. System duty cycle flows or quiescent flow leakage must be sufficient to reject the heat generated by the pump. A fixed displacement pump may safely incorporate an internal drain. Variable delivery pumps with internal drains may be operated at zero flow for short periods of time only (i.e., to set/adjust pressure controls).

5.2 Case Drain Pressure

The pump envelope is directly affected by the internal case pressure level, whereby higher pressures require correspondingly higher design strength in the housing, interfacing joints, and fasteners. The effect of case pressure on the static seals and drive shaft seal members of the pump must be considered and the impact on the unit sealing integrity evaluated.

The case pressure imparts an axial thrust on the drive shaft equal to the product of the shaft seal differential area (area exposed to case pressure minus atmospheric pressure) times the case pressure level. This load is transmitted through the main shaft bearing thereby affecting the life of that bearing.

Present aircraft applications require pumps to be designed for case pressures up to 500 psi (3450 kPa) operating, 1250 psi (8625 kPa) minimum burst. The normal operating level is usually much lower than the level specified for structural integrity.

Back pressure (i.e., the differential pressure between case pressure and pump inlet pressure) has a direct effect on the balance and loading of the pump rotating group. An axial force is exerted on each pumping piston during the suction stroke by a positive case drain back pressure. The piston return mechanism must be designed to withstand this force in addition to the inertia and friction forces inherent to the design.

High back pressure can also cause the pump internal leakage to be "short circuited" back to pump inlet through the piston shoes (slippers) during the suction stroke rather than being vented out the case drain line. This results in a loss of pump heat rejection capability that may cause overheating during low outlet flow operation and the subsequent deterioration of the fluid and pump internal parts.

Ideally, the case drain pressure and inlet pressure should be comparable. The effect of extended operation with high back pressures must be fully understood in the design and use of the pump especially in the low outlet flow modes.

In some designs, an integral relief valve between the pump case and the low pressure inlet port is incorporated. The primary purpose of the relief valve is to relieve a steady-state case overpressurization in order to prevent the rupture of the case housing.

5.2.1 Case Drain Scavenge Pumps

The use of a scavenge pump in the case of the pump is advantageous in systems with potentially persistent high case drain line pressures. It prevents the high case drain line pressures being applied to the rotating group of the pump and assures adequate case drain flow under all conditions to provide optimum cooling of the pump and flow through the system heat exchanger if one is located in the case drain line of the pump.

Care must be exercised in the pump design either to avoid applying the pressure produced by the scavenge pump to the shaft seal of the pump or to design the seal and shaft bearing to accept these pressures and loads.

5.3 Case Drain Line - Direct to Reservoir and Return Line

Case drain lines must be sized to pass the maximum determined case drain flow at pressures at or below the established allowable case pressure for the pump. Factors affecting sizing of the line include the line length to the point of return at the reservoir, and the components in the line (i.e., filter, cooler, valves, etc.). All conditions and modes of pump operation must be considered when establishing the maximum case drain flow. Flow spikes due to the movement and venting of the pump controls during transient operation must also be determined.

A case drain line with an in-line filter and cooler and connected directly to the system reservoir provides the simplest system design and effectively isolates the case drain. Design consideration, such as long line lengths, etc. may dictate connecting the case drain line to the return line or some other low pressure line in the system. The case should be protected from system pressure spikes or back flow by proper valving, check valves, etc. if this potential exists

The system designer should specify system characteristics and that pump testing should be performed in systems incorporating realistic case drain characteristics. The case drain line characteristics should be provided on a plot of flow versus pressure drop between the pump case drain port and the reservoir, or the point of return.

The case drain line often provides the optimum location for the system cooler due to the higher temperature levels and total fluid flow normally associated with this line. Adequate flow for cooling must be maintained to assure proper cooling at all conditions and a filter upstream of the cooler should be utilized to prevent contamination of the cooler which could result in higher case pressures and loss of cooling capacity.

5.4 Case Drain Flow Change

The case drain flow offers the best indicator of the pump's condition. The deterioration of the running/sealing surfaces in a pump will be reflected in an increase in case flow due to increased internal leakages. The detection of this increased flow allows for the early removal of a unit prior to catastrophic failure (i.e., one in which the pump is lost, system hydraulic power lost, and the system contaminated).

The differential pressure indicator incorporated on some filters provides one type of monitoring device. The indicators operate on the pressure drop across the filter element and are intended as fairly crude monitors. They do not differentiate between pressure drop due to increased flow, a dirty element, or high fluid viscosity at low temperatures.

Whatever type monitoring device is used, it is mandatory that it always be checked at the same operating conditions (pump speed, pressure, temperature, and flow). This is necessary to eliminate changes due to varying system pressure, pump speeds, and fluid temperatures.

5.5 Parallel Pump Installations

Those hydraulic systems that utilize parallel pumps where the pump case drains are connected to a common drain line should be designed to prevent fluid backflow between units as backflow may flush contaminants from the drain lines back into the pump case. This could cause a cross contamination between units or reintroduction of contaminants into the pump from which they originated. A backflow condition will occur when the pressure in the common drain line is higher than the case pressure of any one pump. This condition may be caused by a comparatively high case flow from one of several operating pumps or by a system in which one pump is not operating (i.e., the pump is on stand-by).

It is desirable that the common line be sufficiently large and unobstructed so that the pressure in the common line is always lower than the pressure in any of the individual case drain lines. In cases where the line length is not very large, it is desirable to run the individual case drain line directly to the reservoir.

The location of the junction of the drain lines close to a pump case port could also result in backflow into that unit. A check valve installed in each pump case drain line will prevent backflow into the units.

The installation of separate filters for each pump or the incorporation of one filter in the common drain line must be evaluated by the system designer. The installation of separate case drain line filters does not eliminate the need for the check valves. Separate filters will prevent cross-contamination between pumps but will not prevent backwashing of contaminants from the filter into the originating pump.

A filter in the common case drain line from multiple pumps will provide the necessary system protection. However, the failure of one of the pumps may cause high case pressure in all units if the filter element is heavily charged with contaminants. This condition could affect operation of the pumps, compromising the back-up feature of the system.

5.6 Miscellaneous

5.6.1 Pump Connections

Case drain ports are normally marked on the pump housing to prevent the improper assembly of the case drain line. Additional caution is advised to assure proper connection when connecting lines to pumps that have multiple seal drain ports in the same proximity as the case drain port.

5.6.2 Quick Disconnect Couplings

Case drain ports are usually located apart from the pump discharge and supply ports and may not be readily visible. Cautionary measures should be taken to assure that the drain line connection is not overlooked during pump installation. This is especially essential with sealing quick disconnects where the seal must unseat to complete the connection. Ensuing pump operation can result in failure due to an overpressure in the pump case and overheating of the pump.