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VERY LARGE TELESCOPE

ESO Instrument Cabinets: Cabinet Cooling Guide

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Function	Name	Date	Signature
Author	A. Jost		
Job Manager	C. Lucuix		
Releaser	R. Tamai		

☐ This document is under configuration control

ESO, Karl-Schwarzschild-Str. 2, 85748 Garching bei München, Germany

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REVIEWERS

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1 Introduction

1.1 Purpose

This document shows how heat exchanger used in cabinets and other cooling issues can be analysed. The purpose of the document is to show how cooling system can be analysed and verified for applications in Paranal. The basic principle of coolant calculations are explained based on examples.

1.2 Scope

This document allows the reader to analyse cooling requirements for instruments with respect to Paranal environmental conditions.

1.3 Applicable Documents

ID	Document Name	Document Number	Issue	Date
AD1	Environmental Specifications	VLT-SPE-ESO-10000-0004	6	12/11/1997
AD2	List of VLT Service Connection Points	VLT-LIS-ESO-10000-0551	5	29/07/2003
AD3	Requirement for Scientific Instruments on the VLT Unit Telescopes	VLT-SPE-ESO-10000-2723	1	18/03/2005
AD4	Electronics Design Specification	VLT-SPE-ESO-10000-0015	6	8/12/2005
AD5				
AD6				

1.4 Reference Documents

ID	Document Name	Document Number	Issue	Date
RD1	Cabinet cooling	de.wikipedia.org/wiki/Schaltschrankklimatisierung		
RD2	Pressure loss of air ducts	www.klimapartner-berlin.de/default.asp?file=270809.xml		
RD3	Pressure loss calculator for tubes	http://www.schweizer-fn.de/berechnung/stroemung/v2_druckverlust_rohr_rech.htm		
RD4	Pressure loss in rectangular tubes	http://www.westaflex.com		
RD5	Copper tube surface roughness	http://www.schweizer-fn.de/stroemung/rauhigkeit/v2_rauhigkeit.htm		
RD6	Air flow versus pressure	http://www.comairrotron.com/airflow_note.shtml		
RD7	Heat Exchanger Design handbook, T. Kuppan			
RD8				
RD9				

1.5 Acronyms

CFM	Cubic Feet per Minute
MTD	True Mean Temperature Difference
LMTD	Logarithmic Mean Temperature Difference

2 COOLING SPECIFICATIONS FOR THE VLT / PARANAL

2.1 General

The cooling specifications for the VLT are defined in AD1, AD2 & AD3.

The following is a summary of these specifications put together for the convenience of the reader.

2.2 Operating Temperature

Operational Temperature Range: 0°C to 15°C

Humidity: ≤10% @ 20°C, local pressure

2.3 Coolant fluid supply parameters

Coolant water with 33% ethylene glycol

Nominal pressure: 6 bar

Supply differential pressure: ≥0.8bar, ≤ 2bar

Coolant supply temperature: typical 8°C below ambient, min ≥ +4.5°(+5.5°C), max ≤ 7°C, ambient range: 0°C to 15°C (at night)

Note: The coolant supply might deviate from the spec above if the dewpoint + 2.5°C is higher than the coolant temperature derived from above specs. The coolant temperature is then adjusted to dewpoint + 2.5°C!

Coolant parameters:

Specific heat capacity: 3530 J/(kg*K)

Coolant density: 1057 kg/m^3

Viscosity: 6.5mPa*s @-8°C, 3.6mPa*s @7°C

Coolant flow rates per SCP:

Cassegrain Instrument: 12 l/min = 0.72m^3/h

Nasmyth Instrument: 15 l/min = 0.9m^3/h

Max coolant supply ΔT: 8K Twghot – Twgcold

2.4 Coolant air parameters

The density of air dependent on altitude and temperature is given in Table 1. The specific heat capacity of air is shown in Table 3.

2.5 Thermal radiation of instrument surface

The maximum temperature difference between any exposed surface of the instrument (or of any associated equipment) and ambient shall be ≤+1.5°C/-5°C in wind-still condition, with a maxium upwardly convected energy for the instrument and all associated equipment of 150W. The weighting factor to be used for negative energies is 0.3. These thermal requirements are considered to be average values over any 30-minute period.

2.6 Summary worst case conditions @ Paranal

Coolant supply temperature: <7°C → viscosity coolant: 3.6mPa*s @7°C

Ambient temperature: max 15°C

Specific heat capacity: 3530 J/(kg*K)

Coolant density: 1057 kg/m^3

Coolant flow: 12 l/min = 0.72m^3/h ; Cassegrain Instrument:

Coolant supply Δp: ≥0.8bar

Max coolant supply ΔT: 8K

Cp_{air} : 1004.6 [J/kg*K] @ 15°C

$pressure_{@2500m}$= 74101.97 Pa

ρ : 258.143/ T; T [K]

$Q_{radiated}$: <150W over 30min average @ 0°C

Max temperature surface cabinet: T_{amb}-5K ≤ T_{amb} ≤ T_{amb}+1.5K (Tambient worst case 0°C)

3 INTRODUCTION TO CABINET COOLING

3.1 Heat transfer possibilities

The cooling of cabinets with electronic equipment located inside telescopes bears some special challenges. The heat produced by the electronics inside the cabinet must be transferred with a coolant out of the cabinet with minimum radiation of heat to the environment. Heat can be transferred in three ways:

Conduction: The flow of thermal energy through a substance from a higher- to a lower-temperature region. This type of heat transfer is used in ESO cabinet cooling to transfer the heat from inside the cabinet to the outside. Electronics can be directly cooled with cold plates, or the cooling transfer is done via a heat exchanger. The medium used to transfer the heat in cabinets is mostly water or a water/antifreeze mixture.

Convection: Heat transfer in a gas or liquid by the circulation of currents from one region to another. This type of heat transfer is often used to transfer the heat from the electronics via air to the heat exchanger located in the cabinet.

Radiation: Heat transfer in a gas or liquid by the circulation of currents from one region to another. The radiation of heat of cabinets located inside the telescope must be minimized to allow proper operation of the telescope. Radiated heat can cause unwanted air turbulences inside the telescope dome disturbing observations.

3.2 Thermal energy

The thermal energy to cool or heat a medium from temperature T1 to Temperature T2 can be calculated as follows:

$$Q= V*cp* \rho* \Delta T$$

Where

Q is the thermal energy [W]

V is the volume flow [m^3/s]

Cp is the specific heat capacity of the medium [Ws/(kg*K)]

ρ is the density of the medium [kg/m^3]

ΔT =T1 - T2 (Temperatures)

It can be seen from the equations that the heat transfer is proportional dependent on the specific heat capacity cp and the density ρ of the cooling medium.

The equation can be rearranged to calculate the volume flow required to transfer the thermal energy.

$$V= Q/(cp* \rho* \Delta T)$$

3.3 Density of Air

The **density of air**, ρ, is the mass per unit volume of Earth atmosphere and is a useful value in aeronautics and other science. Air density decreases with increasing altitude, as does air pressure. It also changes with variances in temperature or humidity. At sea level and 20 °C, air has a density of approximately 1.2 kg/m^3.

Density of air can be calculated as follows

Where

$$\rho = \frac{p}{R \cdot T}$$

ρ air density [kg/m^3]

R specific gas constant 287.058 [J/(kg*K)]
T absolute Temperature (273.15+ T[°])
Pressure (101325 Pa @0m, 95178 Pa @500m, 74101.97 @2500m, 54192.96 Pa @5000m)

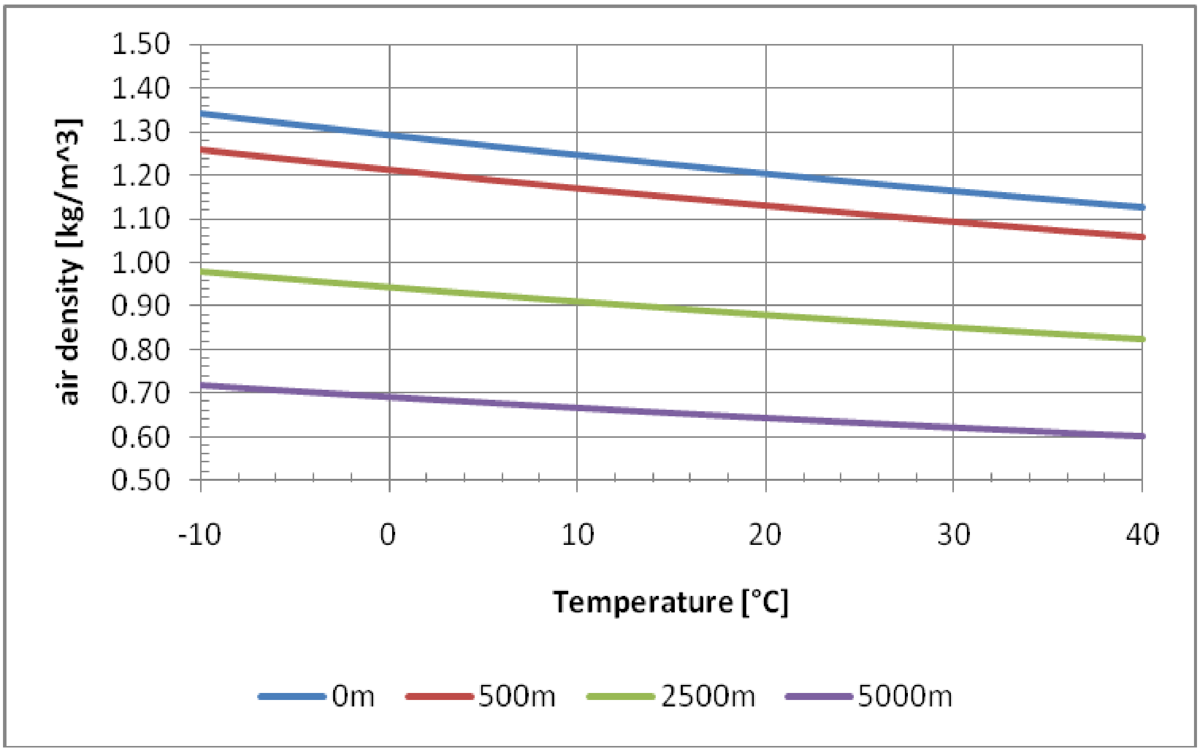


Figure 1 Air density as function of altitude and temperature

The change of air density with humidity is <1% within the temperature range of -10°C to 40°C will be therefore neglected in further calculations.

The figure above and table below show the air density for various altitudes and temperatures. The altitudes shown are

Garching @ 500m above sea
Paranal @ 2500m above sea
Alma @5000m above sea

The air density is 27% lower at Paranal and 47% lower at the ALMA site compared to sea level. The air density decreases with temperature and altitude.

	Altitude	Altitude	Altitude	Altitude
Temperature	0m	500	2500	5000
[°C]	[kg/m^3]	[kg/m^3]	[kg/m^3]	[kg/m^3]
-10	1.341	1.260	0.981	0.717
-8	1.331	1.250	0.974	0.712
-6	1.321	1.241	0.966	0.707
-4	1.311	1.232	0.959	0.701
-2	1.302	1.223	0.952	0.696
0	1.292	1.214	0.945	0.691
2	1.283	1.205	0.938	0.686
4	1.274	1.196	0.931	0.681
6	1.264	1.188	0.925	0.676
8	1.255	1.179	0.918	0.671
10	1.247	1.171	0.912	0.667
12	1.238	1.163	0.905	0.662
14	1.229	1.155	0.899	0.657
16	1.221	1.147	0.893	0.653
18	1.212	1.139	0.887	0.648
20	1.204	1.131	0.881	0.644
22	1.196	1.123	0.875	0.640
24	1.188	1.116	0.869	0.635
26	1.180	1.108	0.863	0.631
28	1.172	1.101	0.857	0.627
30	1.164	1.094	0.852	0.623
32	1.157	1.087	0.846	0.619
34	1.149	1.079	0.840	0.615
36	1.142	1.073	0.835	0.611
38	1.134	1.066	0.830	0.607
40	1.127	1.059	0.824	0.603

Table 1 Density for dry air at different altitudes and temperature

3.4 Water density

Temperature [°C]	Density [kg/m^3]
100	958.4
80	971.8
60	993.2
40	992.2
30	995.6502
25	9999.0479
22	9999.7735
20	9999.2071
15	9999.1026
10	9999.7206
4	999.972

Table 2 Water density for different temperatures

3.5 Specific heat capacity

The heat capacity indicates how much thermal energy ΔQ a physical body can absorb for a change in temperature ΔT. It refers to a specific body, and gives no indication of the amount of substance or composition of the body.

$$C = \frac{\Delta Q}{\Delta T}$$

Where

Q thermal energy

C is the heat capacity of the body [J/K]

ΔT is the change in temperature

Temperature	Heat capacity dry air [J/(kg*K)]	Heat capacity water [J/(kg*K)]
4°C	1004.2	4206
7°C	1004.3	4199
10	1004.4	4193.3
15	1004.6	4186.1
20	1004.8	4181.4
25	1005	4178.7
30	1005.3	4177.5

Table 3 Specific heat capacity of water and air

The table above lists the specific heat capacities for water and air. To simplify calculations the heat capacities can be used with an error of <1% (temperature range 4 to 30°C):

Water = 4206 [J/kg*K] @4°C

Air = 1004.6 [J/kg*K] @ 15°C

The heat capacity is defined in J/K. It can be calculated as

C = specific heat capacity * specific density * Volume

C = J/(kg*K) * kg/m^3 * m^3 = J/K

4 THERMAL RADIATION OF CABINET

4.1 Estimate by Calculation

RD1 shows how to calculate the thermally radiated power by a cabinet surface according to DIN VDE 0660 part 500.

The thermally radiated power by the cabinet can be estimated as follows:

Q= k*A*(Tinside-Toutside)

Q = Thermal radiated power cabinet

k = Thermal transfer conductivity coefficient (steel panel to air incl. Rse + Rsi → k=5.5 W/(m^2*K)

A = effective cabinet surface

Tinside = Temperature inside cabinet [°C]

Toutside = Temperature outside cabinet [°C]

Gehäuse-Aufstellungsart nach DIN VDE 0660 Teil 500	Berechnungsformel der effektiven Schaltschrankoberfläche
Einzelgehäuse allseitig freistehend	$A = 1{,}8 \cdot H \cdot (B+T) + 1{,}4 \cdot B \cdot T$
Einzelgehäuse für Wandaufbau	$A = 1{,}4 \cdot B \cdot (H+T) + 1{,}8 \cdot T \cdot H$
Anfangs-Endgehäuse freistehend	$A = 1{,}4 \cdot T \cdot (H+B) + 1{,}8 \cdot B \cdot H$
Anfangs-Endgehäuse für Wandanbau	$A = 1{,}4 \cdot H \cdot (B+T) + 1{,}4 \cdot B \cdot T$
Mittelgehäuse freistehend	$A = 1{,}8 \cdot B \cdot H + 1{,}4 \cdot B \cdot T + T \cdot H$
Mittelgehäuse für Wandanbau	$A = 1{,}4 \cdot B \cdot (H+T) + T \cdot H$
Mittelgehäuse für Wandanbau mit abgedeckter Dachfläche	$A = 1{,}4 \cdot B \cdot H + 0{,}7 \cdot B \cdot T + T \cdot H$
	B = Schaltschrankbreite H = Schaltschrankhöhe T = Schaltschranktiefe

Table 4 Calculation of the effective surface of cabinets depending free standing or bordering to other cabinets

4.2 Thermal conductivity

The thermal transfer conductivity of a combination of materials can be calculated using the following formula (also known as U-factor used in civil engineering):

k= 1/RT = 1/(Rse+ t_1/λ + t_2/λ + ... + Rsi)

where

k = thermal transfer conductivity [W/(k*m^2)]

RT = total thermal transfer resistance [k*m^2/W]

Rse = transfer resistance external [k*m^2/W]

Rsi = transfer resistance internal [k*m^2/W]; 1/30 W/(k*m^2) for air inside air ducts

4.3 Example: Calculated thermal radiated power for various ΔT for an insulated and not insulated cabinets

The table below shows thermal radiation of an insulated & non insulated cabinet free standing (insulation material armaflex) of size

LxWxD: 2m x 0.6m x 0.6m (revise for 600x800x2000)

calculated applying DIN VDE 0660.

t = thickness of armaflex mat; t=0, cabinet not insulated

ΔT = temperature difference between average air inside cabinet and ambient

t[m]	0	0.01	0.02	0.032
K [W/(K*m^2)]	5.50	2.10	1.30	0.89
ΔT	**Q_r**	**Q_r**	**Q_r**	**Q_r**
[°C]	**[W]**	**[W]**	**[W]**	**[W]**
0	0	0	0	0
5	133	51	31	21
10	265	101	63	43
15	398	152	94	64
20	531	203	125	86
25	663	253	157	107
30	796	304	188	129
35	929	355	219	150
40	1061	405	251	172

Table 5 Thermal radiation cabinet as f(ΔT) and insulation

λ = 0.034 W/(m^2*K); Armaflex @ T= 0-40°C

For black cabinets, the radiated power should be corrected by dividing the $Q_{rblack} = Qr/0.7$.

4.4 Cabinet radiation measured

The thermal power radiated of the cabinet can also be measured by considering the cabinet as a heatsink. The cabinets average outside surface temperature is measured. The thermally radiated power is then calculated by making use of the thermal resistance and the surface of the cabinet.

Assuming cabinet size by HxWxD: 2m x 0.6m x 0.6m and an average surface temperature of 30°C, ambient temperature 20°C the radiated thermal power can be calculated as follows:

A= HxW+W*D, the bottom surface of the cabinet is not taken into account

A= 5.16m^2

Rth = 6K/W for a square surface of 20cmx20m

Rthcabinet = $1/(\text{Acabinet}/A_{0.2x0.2})*1/\text{Rth}) = 0.046511$ k/W

Q = (Tsurface-Tambient) / Rthcabinet

Q = 215W

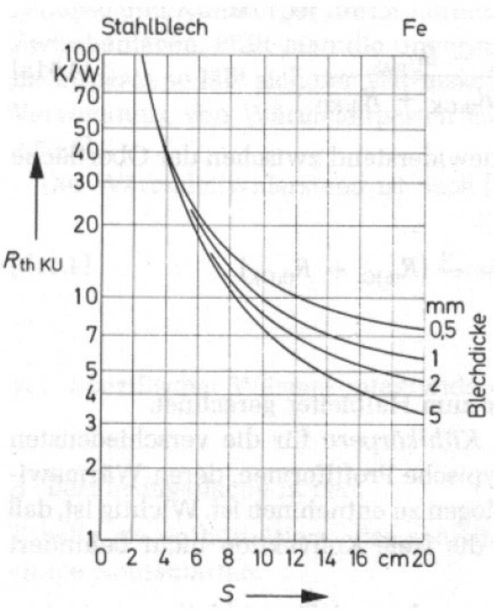
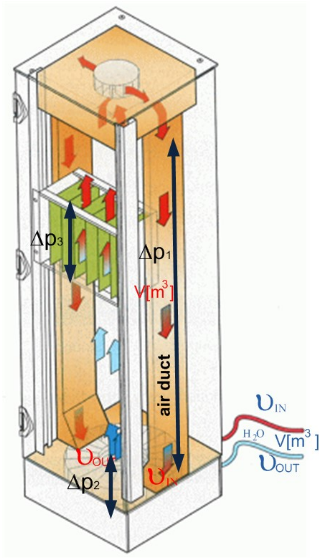


Figure 2 Rth of steel panels

5 HEAT CONVECTION: VOLUME FLOW & PRESSURE

5.1 General



The thermal load inside an electric cabinet is usually cooled by air. Therefore the mass flow of air passing the thermal load plays a significant role in cabinet cooling. The air flow inside the cabinets is usually enforced by fans. The air volume flow provided by a particular fan depends on the system pressure loss of the system to be cooled and the fan speed. The system pressure loss inside an electronic cabinet is the sum of all pressure losses inside the air channel of the cabinet. Each obstacle in the air flow channel produces a loss in pressure – electronic components, heat exchanger, air duct etc.

The following sections have been copied from RD6 for convenience.

5.2 Air volume flow

Before a fan can be specified, the airflow required to dissipate the heat generated has to be approximated. Both the amount of heat to be dissipated and the density of the air must be known.

The basic heat transfer equation is:

$Q = cp * w * \Delta T$

Where

Q amount of heat transferred [W]

Cp specific heat of air ~1004.6 [Ws/(kg*K)]

ΔT temperature change [K]

w mass flow [m/s] $= V * \rho$

where

ρ specific density of air [kg/m^3]
V Volume flow [m^3/h] (= v*A)

$Q = cp * \rho * V * \Delta T \rightarrow V = Q/(cp * \rho * \Delta T)$

$V_{air0m} = 2.9 * W/\Delta T$ m^3/h: $T_{density}$=15°C sea level

This yields a rough estimate of the airflow needed to dissipate a given amount of heat at sea level. It should be noted that the mass of air, not its volume, governs the amount of cooling.

$V_{air2500m} = 4 * W/\Delta T$ m^3/h: $T_{density}$=15°C @ 2500m

$\rightarrow$ $V_{2500m}/ V_{0m} = 1.4$ m^3/h

The air volume flow at an altitude of 2500m must be 1.4 time higher to provide the same heat transfer as at sea level.

5.3 Volume flow tables for water, water-glycol and air

The following tables below show the volume flow of air and water with the associated temperature difference of the medium for various thermal loads. Important temperature differences are highlighted in various colours.

Water	
cpwg	ρwg
4193.3	999.7026

		Vwater	
	[m^3]	[m^3]	[m^3]
	0.9	0.5	0.3
Q			
[W]	ΔTw [°C]	ΔTw [°C]	ΔTw [°C]
5000	4.8	8.6	14.3
4750	4.5	8.2	13.6
4500	4.3	7.7	12.9
4250	4.1	7.3	12.2
4000	3.8	6.9	11.5
3750	3.6	6.4	10.7
3500	3.3	6.0	10.0
3250	3.1	5.6	9.3
3000	2.9	5.2	8.6
2750	2.6	4.7	7.9
2500	2.4	4.3	7.2
2250	2.1	3.9	6.4
2000	1.9	3.4	5.7
1750	1.7	3.0	5.0
1500	1.4	2.6	4.3
1250	1.2	2.1	3.6
1000	1.0	1.7	2.9
750	0.7	1.3	2.1
500	0.5	0.9	1.4
250	0.2	0.4	0.7

Air [20°C]	
cpair	ρair
1004.8	1.204

		Vair		
	[m^3]	[m^3]	[m^3]	[m^3]
	450	600	900	2000
Q				
[W]	ΔTair [°C]	ΔTair [°C]	ΔTair [°C]	ΔTair [°C]
5000	33.1	24.8	16.5	7.4
4750	31.4	23.6	15.7	7.1
4500	29.8	22.3	14.9	6.7
4250	28.1	21.1	14.1	6.3
4000	26.5	19.8	13.2	6.0
3750	24.8	18.6	12.4	5.6
3500	23.1	17.4	11.6	5.2
3250	21.5	16.1	10.7	4.8
3000	19.8	14.9	9.9	4.5
2750	18.2	13.6	9.1	4.1
2500	16.5	12.4	8.3	3.7
2250	14.9	11.2	7.4	3.3
2000	13.2	9.9	6.6	3.0
1750	11.6	8.7	5.8	2.6
1500	9.9	7.4	5.0	2.2
1250	8.3	6.2	4.1	1.9
1000	6.6	5.0	3.3	1.5
750	5.0	3.7	2.5	1.1
500	3.3	2.5	1.7	0.7
250	1.7	1.2	0.8	0.4

Water /Glycol	
cpwg	ρwg
3530	1057

		Vwater	
	[m^3]	[m^3]	[m^3]
	0.9	0.5	0.3
Q			
[W]	ΔTw [°C]	ΔTw [°C]	ΔTw [°C]
5000	5.4	9.6	16.1
4750	5.1	9.2	15.3
4500	4.8	8.7	14.5
4250	4.6	8.2	13.7
4000	4.3	7.7	12.9
3750	4.0	7.2	12.1
3500	3.8	6.8	11.3
3250	3.5	6.3	10.5
3000	3.2	5.8	9.6
2750	2.9	5.3	8.8
2500	2.7	4.8	8.0
2250	2.4	4.3	7.2
2000	2.1	3.9	6.4
1750	1.9	3.4	5.6
1500	1.6	2.9	4.8
1250	1.3	2.4	4.0
1000	1.1	1.9	3.2
750	0.8	1.4	2.4
500	0.5	1.0	1.6
250	0.3	0.5	0.8

Air [20°C]	
cpair	ρair
1004.8	0.881

		Vair		
	[m^3]	[m^3]	[m^3]	[m^3]
	450	600	900	2000
Q				
[W]	ΔTair [°C]	ΔTair [°C]	ΔTair [°C]	ΔTair [°C]
5000	45.2	33.9	22.6	10.2
4750	42.9	32.2	21.5	9.7
4500	40.7	30.5	20.3	9.2
4250	38.4	28.8	19.2	8.6
4000	36.1	27.1	18.1	8.1
3750	33.9	25.4	16.9	7.6
3500	31.6	23.7	15.8	7.1
3250	29.4	22.0	14.7	6.6
3000	27.1	20.3	13.6	6.1
2750	24.9	18.6	12.4	5.6
2500	22.6	16.9	11.3	5.1
2250	20.3	15.3	10.2	4.6
2000	18.1	13.6	9.0	4.1
1750	15.8	11.9	7.9	3.6
1500	13.6	10.2	6.8	3.1
1250	11.3	8.5	5.6	2.5
1000	9.0	6.8	4.5	2.0
750	6.8	5.1	3.4	1.5
500	4.5	3.4	2.3	1.0
250	2.3	1.7	1.1	0.5

0-4°C 4-8°C 0-5°C 5-10°C 10-15°C 0-4°C 4-8°C 0-5°C 5-10°C 10-15°C

Table 6 Left- Water as coolant, air density sea level; Right: Water/Glycol as coolant, air density @2500m

Note: The temperature rise of water-glycol is specified for VLT applications to of ΔT_{wg}=0-8°C. The air temperature rise for VLT application is limited at the lower side by the max ambient temperature - <15°C, on the upper side by the max temperature inside the cabinet - <30°C (Electronic limit). Therefore ΔT_{air}=0-15°C. The air temperature range of 0-5°C is only of interest if very high airflow can be guaranteed or the load is very low. For standard cabinets (0.6mx0.6mx2m) a volume air flow in the range of 450m^3 to 900m^3 is achievable in practice. Therefore the power dissipation in those cabinets is limited to the range of 1.5kW – 3kW max for a temperature increase of the air <15°C.revise for 40deg cabinet

5.4 System impedance

After the airflow has been determined, the amount of resistance to it must be found. This resistance to flow is referred to as system impedance and is expressed in static pressure as a function of volume flow. A typical system impedance curve which most electronic equipment follows, is called the "square law," which means that static pressure changes as a square to the system impedance and is expressed in static pressure as a function of volume flow. Figure 3 describes typical impedance curves. For most forced air cooling application, the system curve is calculated by:

P= K*r* Q^n

Where

P static pressure

K load factor

r fluid density

V Volume flow

n constant; let n=2; approximating a turbulent system

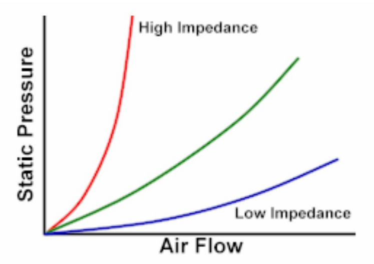


Figure 3 System impedance vs air flow

Static pressure through complex systems cannot be easily arrived at by calculation. In any system, measurement of the static pressure will provide the most accurate result.

5.5 System flow

Once the volume of air and the static pressure of the system to be cooled are known, it is possible to specify a fan. The governing principle in fan selection is that any given fan can only deliver one flow at one pressure in a given system. Figure 4 shows a typical fan pressure versus flow curve along with what is considered the normal operating range of the fan. The fan, in any given system, can only deliver as much air as the system will pass for a given pressure. Thus, before increasing the number of fans in a system, or attempting to increase the air volume flow using a larger fan, the system should be analyzed for possible reduction in the overall resistance to airflow. Other considerations, such as available space and power, noise, reliability, vibration and operating environment should also be brought to bear on fan choice.

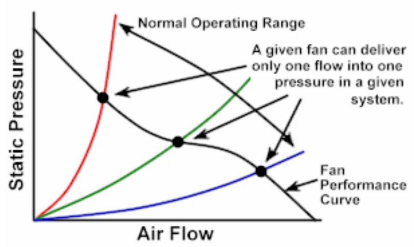


Figure 4 Fan flow over operating range

5.6 Impact of varying system impedance

To demonstrate the impact of system resistance on fan performance, figure 3 shows three typical fans used in the computer industry. A is a 120 CFM fan, B is a 100 CFM fan and C is a 70CFM fan. Line D represents a system impedance within a given designed system. If 50 CFM of air are needed, fan A will meet the need. However, fan A is a high performance, higher noise fan that will likely draw more power and be more costly. If the system impedance could be improved to curve E, then fan B would meet the 50 CFM requirement, with a probable reduction in cost, noise and power draw. And if the system impedance could be optimized to where curve F were representative, then fan C would meet the airflow requirement, at a dramatically lower power, noise and cost level. This would be considered a well-designed system from a forced convection cooling viewpoint. Keeping in mind that a given fan can only deliver a single airflow into a given system impedance, the importance of system design on fan selection is critical. Comair Rotron urges engineers to minimize system impedance where practical, for best performance, noise, power and cost characteristics.

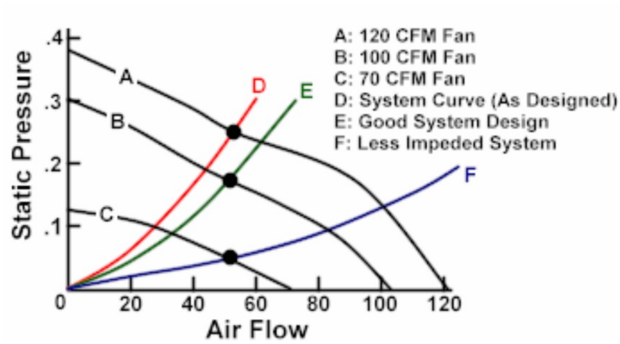


Figure 5 System impedance impact

5.7 Series and Parallel Operation

Combining fans in series of parallel can achieve the desired airflow without greatly increasing the system package size or fan diameter. Parallel operation is defined as having two or more fans blowing together side by side. The performance of two fans in parallel will result in doubling the volume flow, but only at free delivery. As Figure 6 shows, when a system curve is overlaid on the parallel performance curves, the higher the system resistance, the less increase in flow results with parallel fan operation. Thus, this type of application should only be used when the fans can operate in low impedance near free delivery.

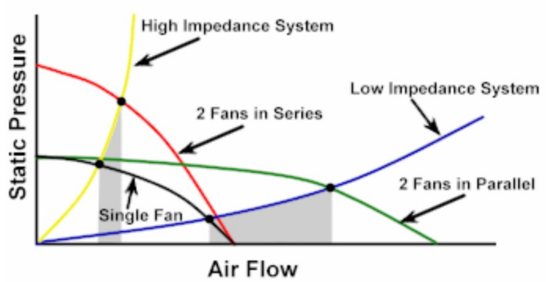


Figure 6 Series vs parallel performance

Series operation can be defined as using multiple fans in a push-pull arrangement. By staging two fans in series, the static pressure capability at a given airflow can be increased, but again, not doubled at every flow point, as Figure 7 displays. In series operation, the best results are achieved in systems with high impedance.

In both series and parallel operation, particularly with multiple fans (5, 6, 7, etc.) certain areas of the combined performance curve will be unstable and should be avoided. This instability is unpredictable and is a function of the fan and motor construction and the operating point. For multiple fan installations, it is strongly recommended to test the system in the laboratory.

5.8 Speed and Density Changes

By using dimensional analysis and fluid dynamic equations, basic fan laws can be derived giving a relationship between airflow, static pressure, horsepower, speed, density and noise. The table below shows the most useful of these fan laws.

Basic Fan Laws

Variable	When Speed Changes	When Density Changes
Air Flow	Varies directly with speed ratio: $CFM_2 = CFM_1\ (RPM_2 / RPM_1)$	Varies directly with density ratio: $CFM_2 = CFM_1\ (r_2 / r_1)$
Pressure	Varies with square of speed ratio: $P_2 = P_1\ (RPM_2 / RPM_1)^2$	Varies directly with density ratio: $P_2 = P_1\ (r_2 / r_1)$
Power	Varies with cube of speed ratio: $HP_2 = HP_1\ (RPM_2 / RPM_1)^3$	Varies directly with density ratio: $HP_2 = HP_1\ (r_2 / r_1)$
Noise	$N_2 = N_1 + 50 \log_{10}(RPM_2 / RPM_1)$	$N_2 = N_1 + 20 \log_{10}(r_2 / r_1)$

Table 7

As an example of the interaction of the fan laws, assume we want to increase airflow out of a fan by 10%. By increasing the fan speed 10%, we will achieve the increased airflow. However, this will require 33% more horsepower from the fan motor. Usually, the fan motor is being fully used and has no extra horsepower capability. Other solutions will have to be considered. The fan laws can be extremely useful in predicting the effect on fan performance and specification when certain operating parameters are changed.

5.9 Density effects on fan performance

Since a fan is a constant volume machine, it will move the same CFM of air no matter what density of the air as seen in Figure 7. However, a fan is not a constant mass flow machine. Therefore, mass flow changes as the density changes. This becomes important when equipment must operate at various altitudes. The mass flow is directly proportional to density change, while the volume flow (CFM) remains constant. As air density decreases, mass flow decreases and the effective cooling will diminish proportionately. Therefore, equivalent mass flow is needed for equivalent cooling, or the volume flow (CFM) required at altitude (low density) will be greater than what required at sea level to obtain equivalent heat dissipation

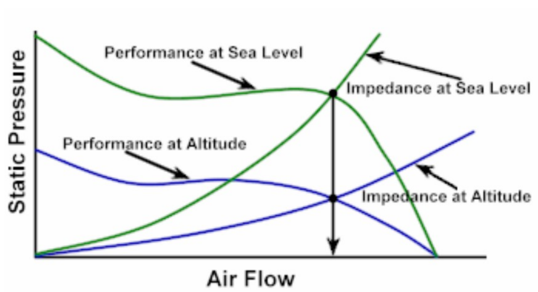


Figure 7 Density effects on fan performance

5.10 Pressure loss in air ducts

The pressure loss in air ducts can either be calculated or taken from charts. Some examples are given in the following sections.

5.10.1 Pressure loss for round air ducts

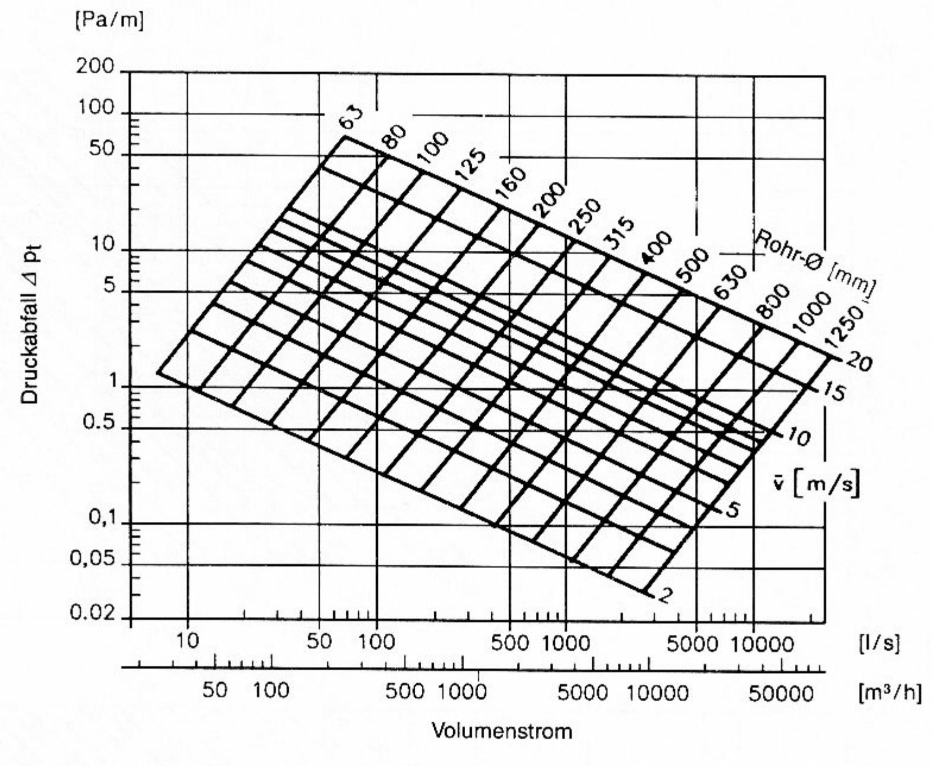


Figure 8 Pressure loss of round air ducts as function of diameter, air volume flow, and velocity [RD2]

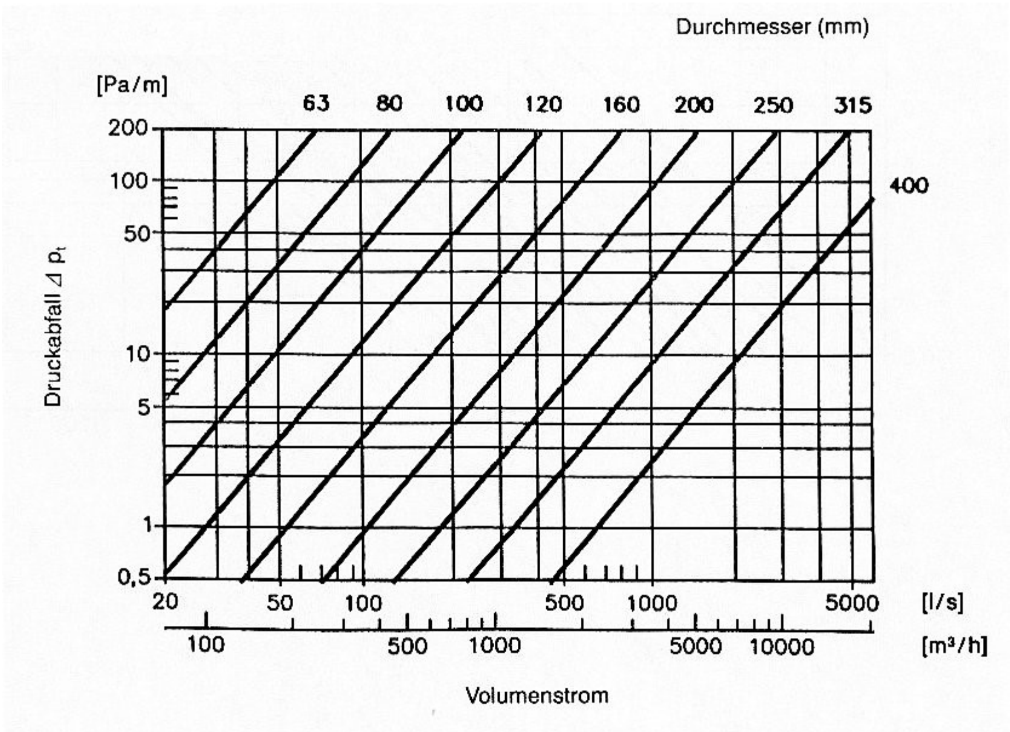


Figure 9 Pressure loss of round air ducts as function of diameter and air volume flow [RD2]

The figure above shows the pressure loss in a round air duct as function of diameter of the air duct and the air volume flow. The pressure is given in Pa/m.

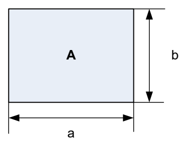
The pressure loss for a tube with a length of 2m, diameter of 160mm, a volume flow of 800m^3 and a speed of 10m/s results in a pressure drop of 6Pa/m →12Pa over 2m.

The pressure loss in air ducts can be calculated or taken from charts as shown in Figure 8 & Figure 10 for air ducts according to [RD4].

The pressure loss can also be calculated with the following formulae:

1) For rectangular air ducts calculate the equivalent hydraulic diameter (dh=d in case of round tubes):

$$d_h = 2*a*b/(a+b) \quad [m]$$



2) Calculate the speed of the volume flow of air in the duct

$$w = V/(A*3600) \quad [s]$$

Where

w= speed of air inside the duct [m/s]

A= cross section of the square duct [m^2]

V= Volume flow rate [m^3/h]

3) Determine the Reynold-number for Re

$$Re = w*d_h/v$$

Where

v= 15*10°-6 m^2/s for air

4) Calc tube friction coefficient

$\lambda = 0.22 / Re^{0.2}$

5) Calculate pressure drop

$$\Delta p = \lambda * L * \rho\, w^2 / (2*d_h) \quad [Pa]$$

Where

L = tube length

ρ = air density (1.225 kg/m^3 @ 15°C and altitude 0m)

Note: The pressure drop decreases with increase of altitude and/or increase of temperature increase as the air density is decreases.

5.10.2 Squared air ducts

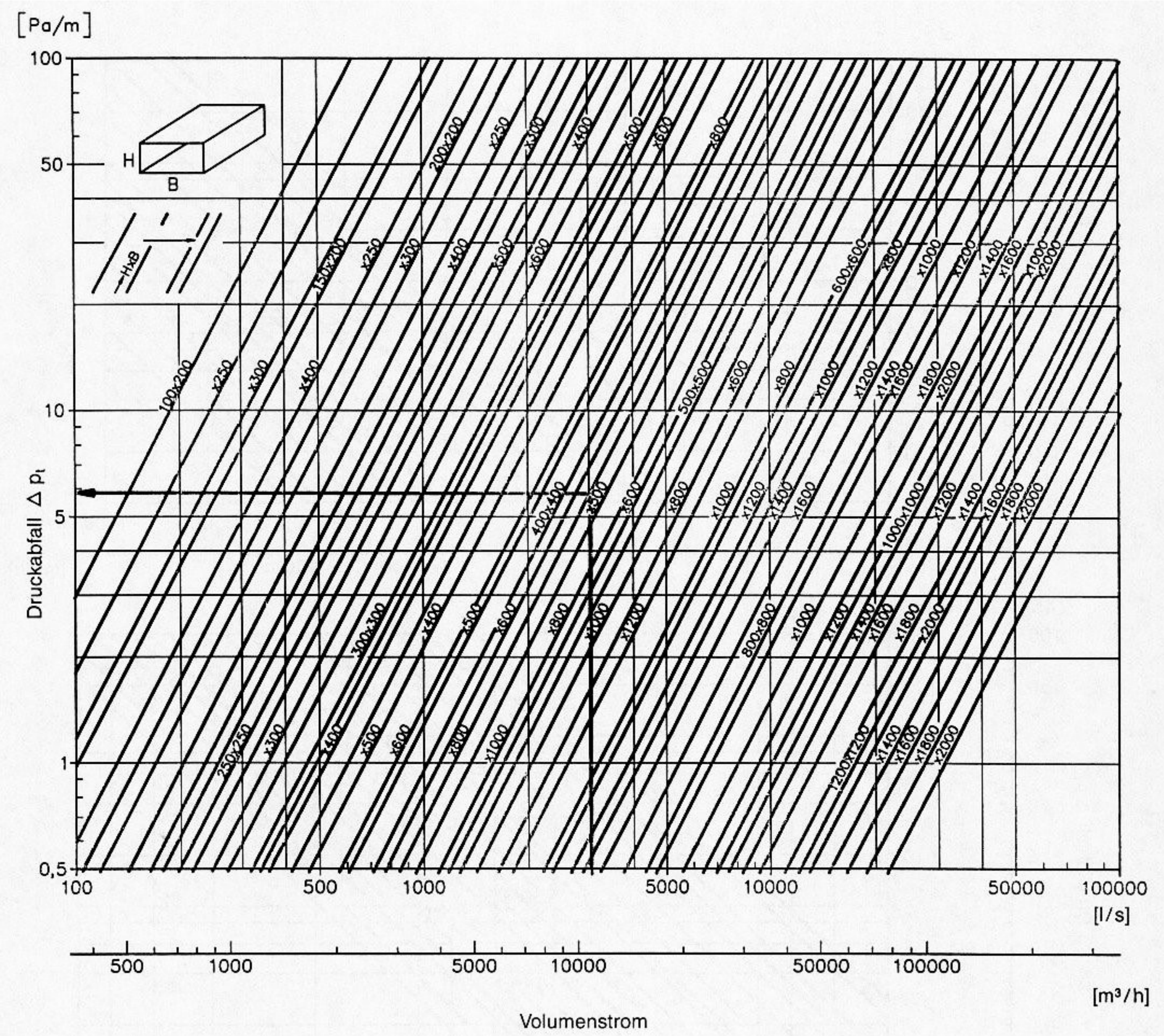


Figure 10 Pressure loss for rectangular air ducts, H & B are in mm [RD2]

5.10.3 Pressure loss in bends of air ducts

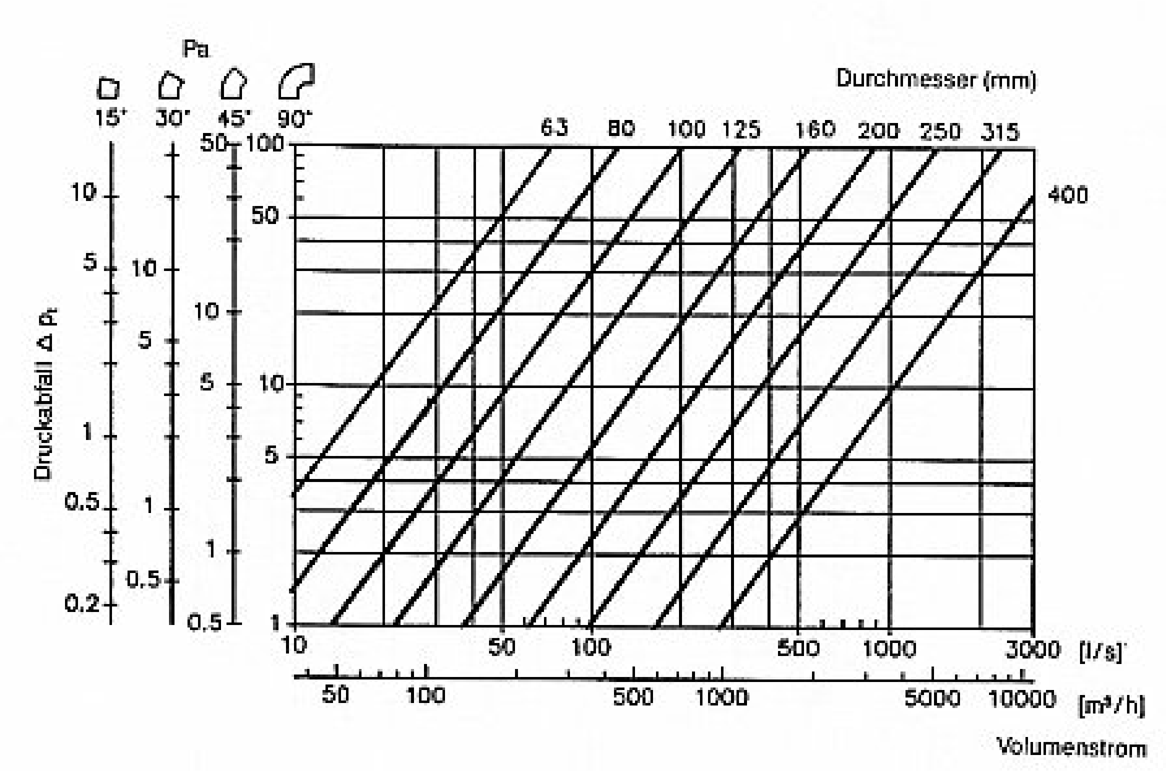


Figure 11 Pressure loss in bends as function of duct diameter and volume flow up to d=400mm[RD2]

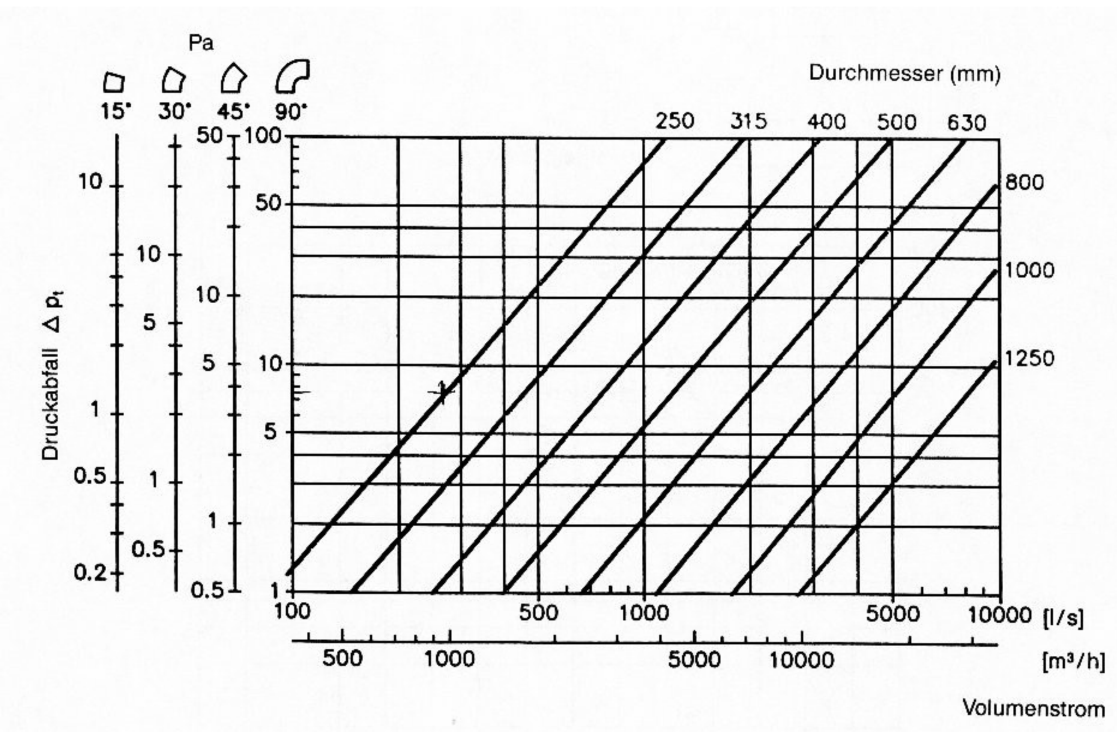


Figure 12 Pressure loss in bends as function of duct diameter and volume flow up to d=1250mm [RD2]

5.10.4 Pressure loss in air duct junctions

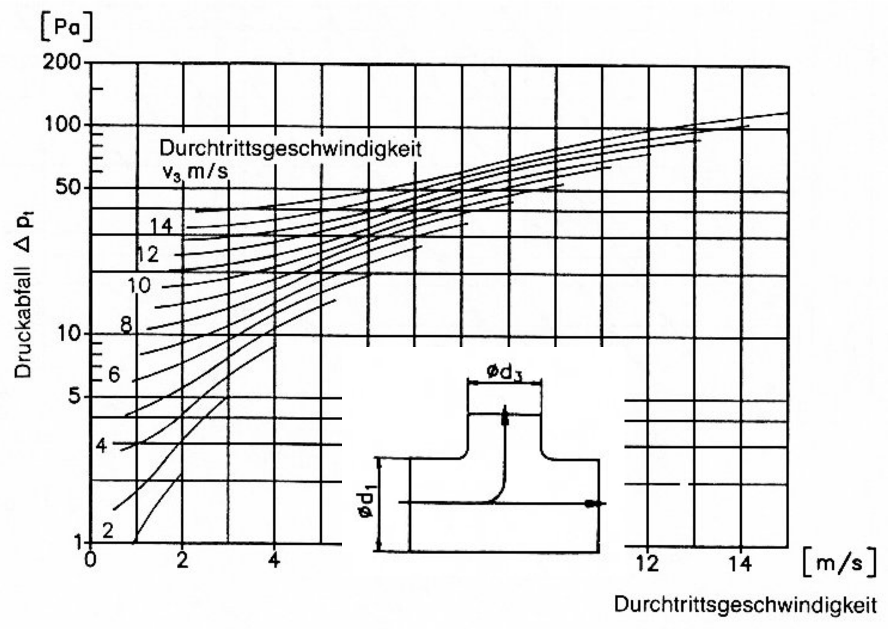


Figure 13 Pressure loss in junctions - split 90° [RD2]

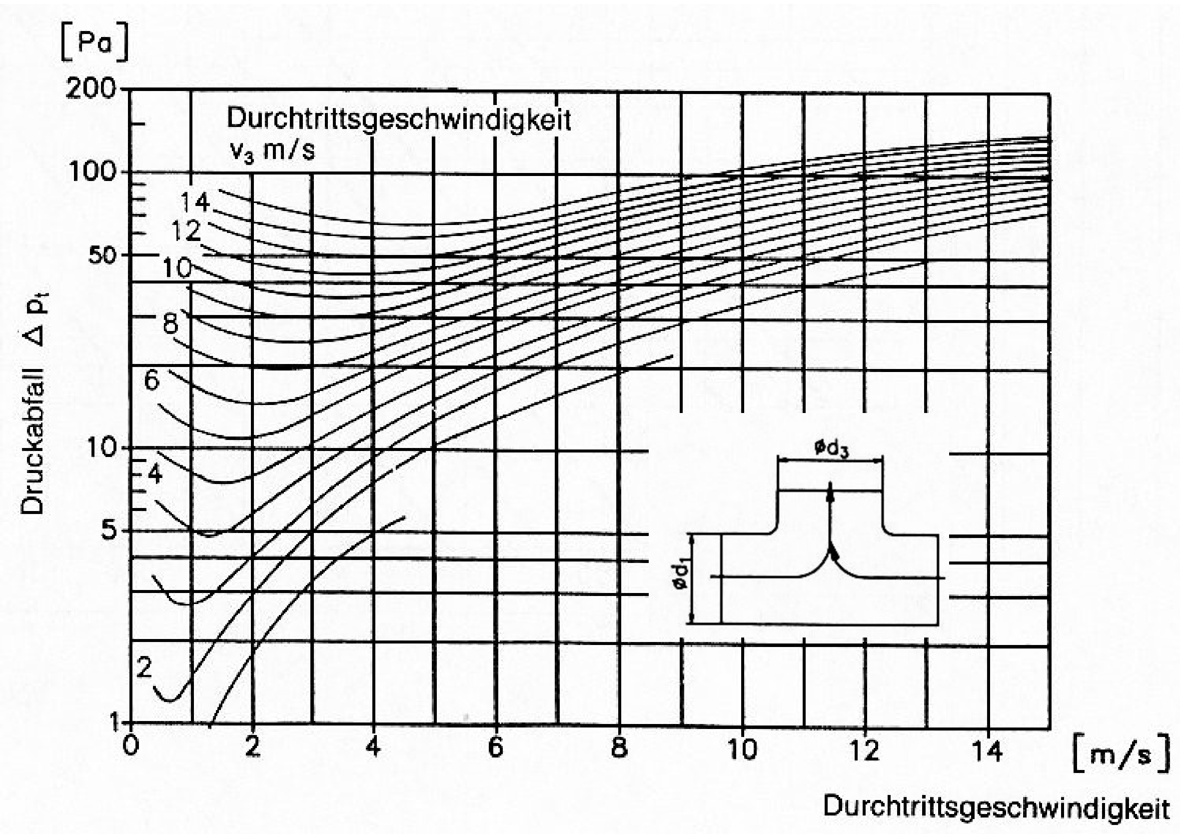


Figure 14 Pressure loss in junctions - combine 90° [RD2]

6 WATER FLOW

6.1 Pressure loss in copper ducts for coolant flow

Tube surface roughness copper/plastic tubes: λ= 0.01-0.03mm [RD5].

$$\Delta p = \lambda * L * \rho * v^2 / (2*d_i) \quad [Pa]$$

Where

λ surface roughness tube
L length tube [m]
ρ specific density media [kg/m^3]
v flow velocity [m/s]
di diameter inside tube [m]

6.2 Pressure loss in bended rigid round tubes

The additional loss due to bends in round ducts can be calculated by calculating an "equivalent length" for the bend. The pressure loss can then be derived using the formula above for the extended length. Values of the equivalent length (Le) may be determined using the internal diameter (D) of the hose in the following relationship:

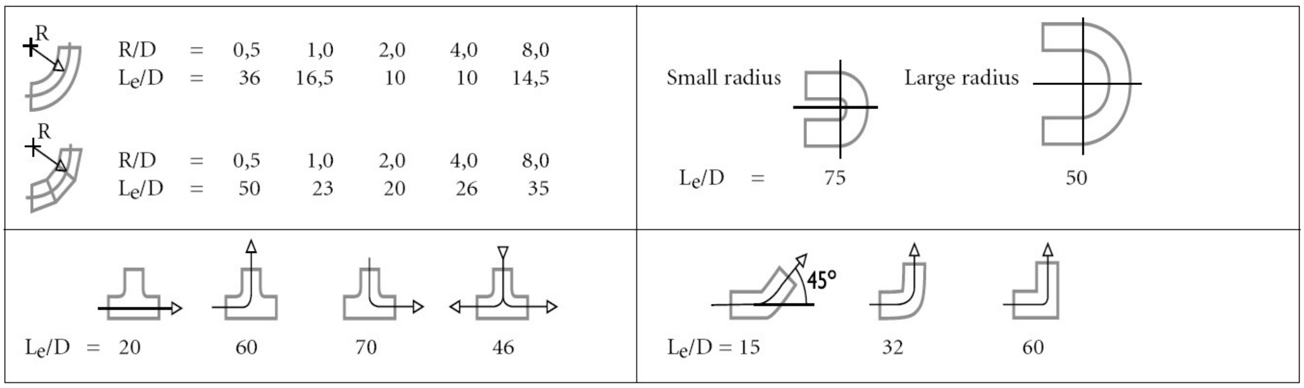


Figure 15 Chart for bends in ducts

7 HEAT TRANSFER IN HEAT EXCHANGERS

The heat transfer in heat exchanger can be calculated with the following equation:

Q= U*A*F* LMTD

Where

Q = heat transfer rate [W]

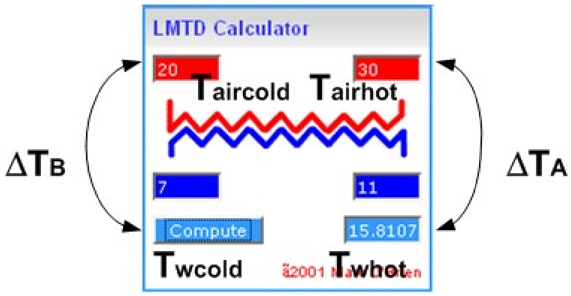
U = overall heat transfer coefficient [W/(m^2*K)]

A = heat transfer surface area [m^2]

F = ratio of the true mean temperature difference (MTD) to the logarithmic mean temperature difference (LMTD)

LMTD = log mean temperature difference [K]

LMTD = $(\Delta T_A - \Delta T_B)/(\ln(\Delta T_A/\Delta T_B))$



F corrects the LMTD which is valid for the counterflow heat exchanger for the MTD of a cross-counterflow heat exchanger which is used for watercooled air heat exchanger (as used for cabinet cooling).

For a fixed heat exchanger surface and architecture, the LMTD is proportional to the heat transfer rate.

Assuming a max temperature T_{airhot} of = 25°C, and a min. T_{whot}=T_{wcold} = 7°C → ΔTA = 18°C. T_{airhot} – $T_{aircold}$ can be assumed to be at least 5°C or higher → ΔTB = $T_{aircold}$ - T_{wcold} =(25°C-5°C)-7°C= 13°C. The figure below shows the LMTD range for these temperatures. The resulting LMTD is 1…15°C. This figure allows a first estimation of min and max surface required for the heat exchanger, or allows an estimate on the power limit of a given heat exchanger using the formula given above.

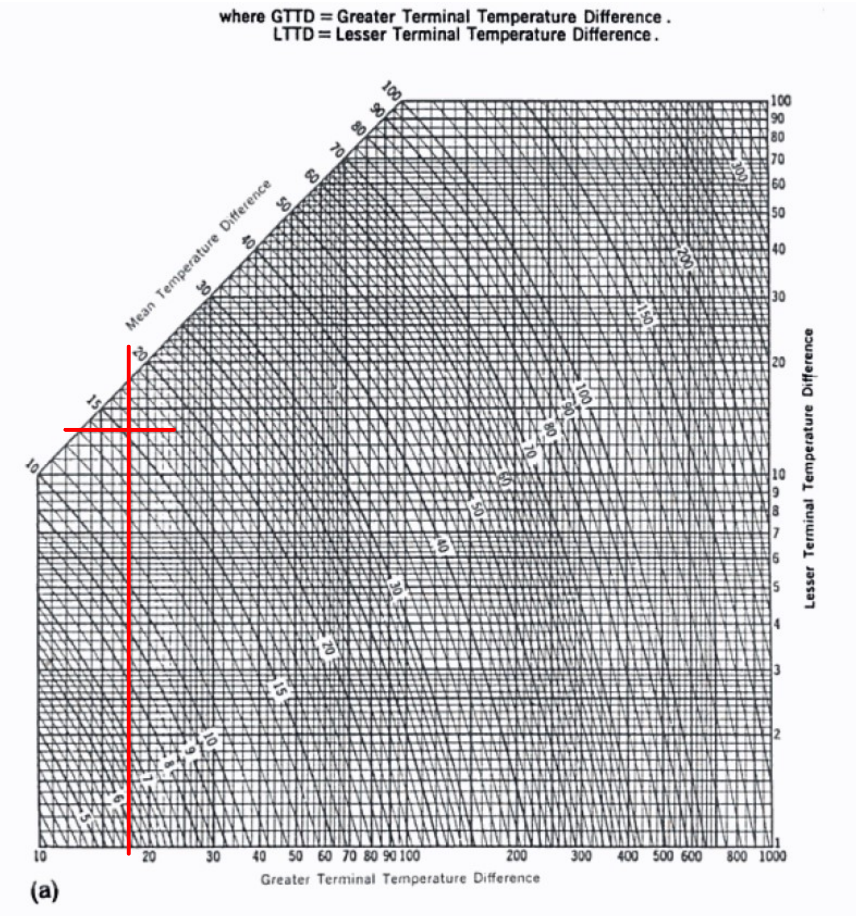


Figure 16 LMTD Chart

For a given heat exchanger U, F and A is a constant figure. Therefore Q is proportional with ΔT_{LMTD}. Q increases linear with LMTD. The lower LMTD, the higher is the thermal load the heat exchanger can extract from the cabinet. Figure 16shows LMTD for various temperatures of the air flowing into the heat exchanger, while the air temperature leaving the heat exchanger is kept fixed. The ΔT of the coolant is also kept fixed. In practice this can be achieved by controlling the volume flow of the coolant.

The optimum for VLT applications would be to regulate the temperature of the air leaving the heat exchanger to

$T_{ambient}$ – 5K → worst case would be $T_{ambient}$=15°C → Tair Out = 10°C (as shown in ..)Comparing the figures above for Temp air Out= 15°C and Temp air Out= 10°C shows that LMTD reduces from 8 to 5.1. The reduction of thermal load extracted reduces therefore by factor 1.57 if the coolant flow is kept constant for both cases.

To achieve the same Q either the surface of the heat exchanger must be increased by factor 1.57 or the coolant flow must be increased accordingly.

For insulated cabinet, the maximum Q extracted for an ideal heat exchanger can be found as follows:

Max temperature air in = 30°C (limited by electronics inside the cabinet)

Min temperature coolant in: 7°C

Max water flow: 15l/min = 0.9m^3

$Q_w = V_w * cp_w * \rho_w * (T_wout-T_win) / 3600$
$Q_{air}= V_{air} * cp_{air} * \rho_{air} * (T_{air}in- T_{air}out) / 3600$

$V_{air}= Q_{air} /(cp_{air} * \rho_{air} * (T_{air}in - T_{air}out))$

Condition: $T_{air}out \geq T_wout$; Twout max $\leq$ Twin +8°K (spec VLT)

$T_{air}out = T_wout + x$; x= (0…8)

$T_wout = T_win + Q *3600 / (V_w * cp_w * \rho_w)$
$V_{air}= Q*3600 /(cp_{air} * \rho_{air} * (T_{air}in - (T_win + Q / (V_w * cp_w * \rho_w)))$

This equation allows to calculate the V_{air} as function of Q, $T_{air}in$ and T_win.

The temperatures can also be calculated for a know heat exchanger base on:
$Q= U*A*F*\Delta T_{LMTD}$

$\Delta T_{LMTD} = (\Delta T_A-\Delta T_B)/(\ln(\Delta T_A/\Delta T_B))$

$Q_{air}= V_{air} * cp_{air} * \rho_{air} * (T_{air}in- T_{air}out) / 3600$

$T_{air}out = T_{air}in - Q_{air} / (V_{air} * cp_{air} * \rho_{air})$

$T_wout = T_win + Q / (V_w * cp_w * \rho_w)$

$\Delta T_{LMTD} = T_{air}in - (T_win + Q *3600 / (V_w * cp_w * \rho_w)) – (T_{air}in - Q_{air} *3600 / (V_{air} * cp_{air} * \rho_{air}) – T_win) / \ln ((T_{air}in - (T_win + Q*3600 / (V_w * cp_w * \rho_w)) / (T_{air}in - Q_{air} *3600 / (V_{air} * cp_{air} * \rho_{air}) – T_win))$

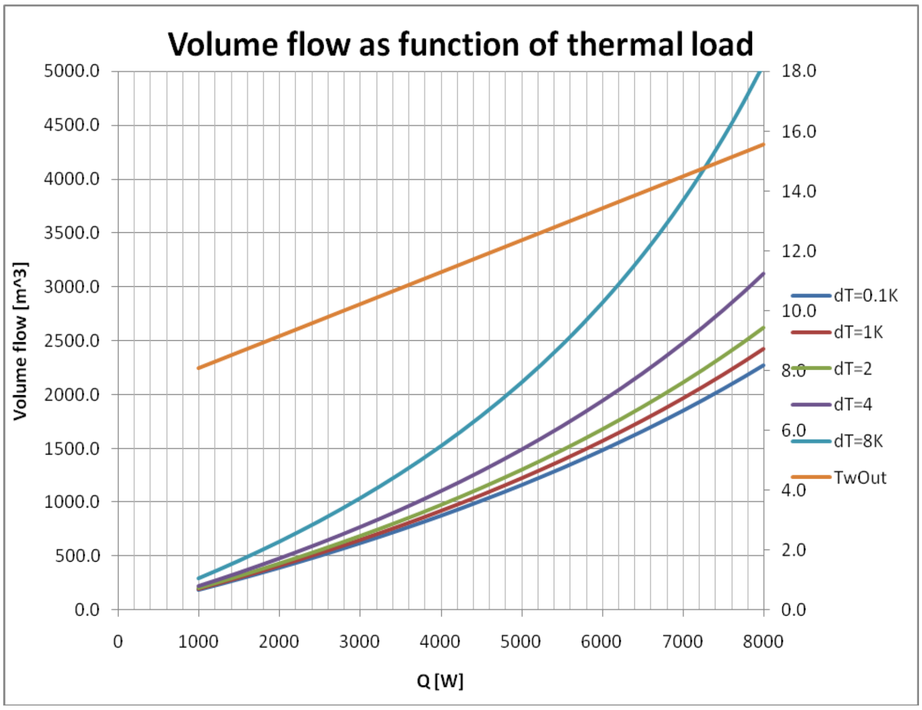


Figure 17

The various dT curves show $T_{air}out$ as function of TwOut, where $T_{air}out$ = TwOut+dT.

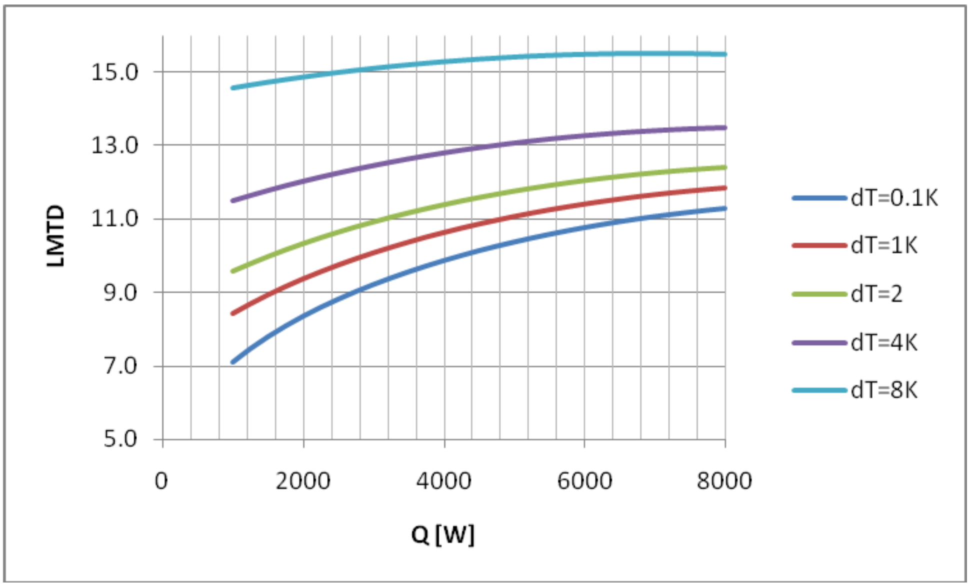


Figure 18 LMTD as function of Q

Example:

Schroff LX3

Q=3kW at LMTD = 10.72K (T_A=8K, T_B=14K)

A= 0.019m * 0.175m * 2*420

A= 2.8m^2

U = 3000W /(2.8m^2*10.72K)

U = 100

8 CALCULATED THERMAL LOAD TABLE

8.1 General

The following pages show tables calculated for various cabinet temperatures and volume flows. The tables are based on the following conditions.

V_{air}= Q*3600 /(cp_{air}* ρ_{air} * (T_{airhot} - (T_{wgcold} + Q / (V_{wg} * cp_{wg}* ρ_{wg})))
T_{wghot} = T_{wgcold} + Q *3600 / (V_w * cp_{wg}* ρ_{wg})
$T_{aircold}$ =T_{airhot} - Q *3600 / V_{air}* cp_{air}* ρ_{air}
T_{airhot} –T_{wghot} = 0…8

LMTD= (T_{airhot} - T_{wghot}) - ($T_{aircold}$ –T_{wgcold}) / ln[(T_{airhot} - T_{wghot}) - ($T_{aircold}$ –T_{wgcold})]

A = Q/(U*F) *LMTD ; for F*U = 88 W/(m^2*K)

The required surface of the heat exchanger is calculated based on the formula above for the thermal load based on a fixed FU value. This FU value has been derived for the heat exchanger delivered with the Schroff cabinet LX3. The value is calculated based on measurements provided by Schroff. The FU value for other heat exchanger will be different from this one. Nevertheless, the calculated surface gives a figure of merit to compare the required surface for different parameters.

Air		Water Glycol		Cooler
Cp air	ρ air	Cp cool	ρ cool	FU
1004.8	0.881	3530	1057	127

Table 8 Cooling Conditions (altitude 2500m)

The top of each page shows the reference table for the following conditions:

Tairhot	Twgcold	Twghot -Tairhot	Vwg
[°C]	[°C]	[°C]	m^3/h
16.5	7.0	0.1 to 8	0.9

Table 9 Reference conditions

8.2 Change of coolant volume flow from 0.9m^3/h to 0.5m^3/h

Table 10 Coolant flow rate 0.5m^3/h

8.3 Change of coolant inlet temperature from 7°C to 11°C

Table 11 Coolant inlet temperature 11°C

8.4 Change of inlet air temperature from 16.5°C to 25°C

T_{airhot}	T_{wgcold}	T_{wghot} - T_{airhot}		V_{wg}
[°C]	[°C]	[°C]		m^3/h
16.5	7.0	0.1		0.9

					Cooler
Q	V_{air}	T_{wghot}	$T_{aircold}$	LMTD	Surface
[W]	m^3	[°C]	[°C]	[°C]	[m^2]
5000	5033	12.4	12.5	4.8	8.3
4750	4484	12.1	12.2	4.8	7.8
4500	3999	11.8	11.9	4.8	7.4
4250	3568	11.6	11.7	4.8	7.0
4000	3182	11.3	11.4	4.8	6.6
3750	2835	11.0	11.1	4.8	6.2
3500	2520	10.8	10.9	4.7	5.8
3250	2234	10.5	10.6	4.7	5.4
3000	1973	10.2	10.3	4.6	5.1
2750	1733	9.9	10.0	4.6	4.7
2500	1513	9.7	9.8	4.5	4.4
2250	1309	9.4	9.5	4.4	4.0
2000	1121	9.1	9.2	4.3	3.7
1750	946	8.9	9.0	4.2	3.3
1500	783	8.6	8.7	4.0	2.9
1250	631	8.3	8.4	3.9	2.5
1000	488	8.1	8.2	3.7	2.1
750	355	7.8	7.9	3.4	1.7
500	229	7.5	7.6	3.1	1.3
250	111	7.3	7.4	2.8	0.7
100	44	7.1	7.2	2.4	0.3

T_{airhot}	T_{wgcold}	T_{wghot} - T_{airhot}		V_{wg}
[°C]	[°C]	[°C]		m^3/h
16.5	7.0	1.0		0.9

					Cooler
Q	V_{air}	T_{wghot}	$T_{aircold}$	LMTD	Surface
[W]	m^3	[°C]	[°C]	[°C]	[m^2]
5000	6476	12.4	13.4	5.2	7.6
4750	5668	12.1	13.1	5.2	7.2
4500	4979	11.8	12.8	5.2	6.8
4250	4382	11.6	12.6	5.2	6.4
4000	3862	11.3	12.3	5.2	6.0
3750	3404	11.0	12.0	5.2	5.6
3500	2998	10.8	11.8	5.2	5.3
3250	2635	10.5	11.5	5.2	4.9
3000	2309	10.2	11.2	5.2	4.6
2750	2014	9.9	10.9	5.1	4.2
2500	1747	9.7	10.7	5.1	3.9
2250	1503	9.4	10.4	5.0	3.5
2000	1280	9.1	10.1	5.0	3.2
1750	1074	8.9	9.9	4.9	2.8
1500	885	8.6	9.6	4.8	2.5
1250	710	8.3	9.3	4.7	2.1
1000	547	8.1	9.1	4.5	1.7
750	396	7.8	8.8	4.4	1.3
500	255	7.5	8.5	4.2	0.9
250	124	7.3	8.3	4.0	0.5
100	48	7.1	8.1	3.9	0.2

T_{airhot}	T_{wgcold}	T_{wghot} - T_{airhot}		V_{wg}
[°C]	[°C]	[°C]		m^3/h
16.5	7.0	4.0		0.9

					Cooler
Q	V_{air}	T_{wghot}	$T_{aircold}$	LMTD	Surface
[W]	m^3	[°C]	[°C]	[°C]	[m^2]
5000	145440	12.4	16.4	6.4	6.2
4750	47367	12.1	16.1	6.5	5.8
4500	27078	11.8	15.8	6.5	5.4
4250	18312	11.6	15.6	6.6	5.1
4000	13423	11.3	15.3	6.6	4.7
3750	10305	11.0	15.0	6.7	4.4
3500	8143	10.8	14.8	6.7	4.1
3250	6556	10.5	14.5	6.7	3.8
3000	5342	10.2	14.2	6.7	3.5
2750	4382	9.9	13.9	6.7	3.2
2500	3605	9.7	13.7	6.7	2.9
2250	2963	9.4	13.4	6.7	2.6
2000	2424	9.1	13.1	6.7	2.3
1750	1964	8.9	12.9	6.7	2.1
1500	1567	8.6	12.6	6.7	1.8
1250	1222	8.3	12.3	6.7	1.5
1000	918	8.1	12.1	6.6	1.2
750	650	7.8	11.8	6.6	0.9
500	410	7.5	11.5	6.5	0.6
250	194	7.3	11.3	6.4	0.3
100	75	7.1	11.1	6.4	0.1

T_{airhot}	T_{wgcold}	T_{wghot} - T_{airhot}		V_{wg}
[°C]	[°C]	[°C]		m^3/h
25	7.0	0.1		0.9

					Cooler
Q	V_{air}	T_{wghot}	$T_{aircold}$	LMTD	Surface
[W]	m^3	[°C]	[°C]	[°C]	[m^2]
5000	1622	12.4	12.5	8.6	4.6
4750	1508	12.1	12.2	8.5	4.4
4500	1400	11.8	11.9	8.4	4.2
4250	1295	11.6	11.7	8.3	4.0
4000	1195	11.3	11.4	8.2	3.8
3750	1099	11.0	11.1	8.1	3.7
3500	1006	10.8	10.9	7.9	3.5
3250	917	10.5	10.6	7.8	3.3
3000	831	10.2	10.3	7.7	3.1
2750	748	9.9	10.0	7.5	2.9
2500	668	9.7	9.8	7.3	2.7
2250	591	9.4	9.5	7.2	2.5
2000	516	9.1	9.2	7.0	2.3
1750	444	8.9	9.0	6.7	2.0
1500	374	8.6	8.7	6.5	1.8
1250	307	8.3	8.4	6.2	1.6
1000	242	8.1	8.2	5.9	1.3
750	178	7.8	7.9	5.5	1.1
500	117	7.5	7.6	5.1	0.8
250	58	7.3	7.4	4.5	0.4
100	23	7.1	7.2	4.0	0.2

T_{airhot}	T_{wgcold}	T_{wghot} - T_{airhot}		V_{wg}
[°C]	[°C]	[°C]		m^3/h
25	7.0	1.0		0.9

					Cooler
Q	V_{air}	T_{wghot}	$T_{aircold}$	LMTD	Surface
[W]	m^3	[°C]	[°C]	[°C]	[m^2]
5000	1747	12.4	13.4	9.1	4.3
4750	1622	12.1	13.1	9.1	4.1
4500	1503	11.8	12.8	9.0	3.9
4250	1389	11.6	12.6	8.9	3.7
4000	1280	11.3	12.3	8.8	3.6
3750	1175	11.0	12.0	8.7	3.4
3500	1074	10.8	11.8	8.6	3.2
3250	978	10.5	11.5	8.5	3.0
3000	885	10.2	11.2	8.4	2.8
2750	796	9.9	10.9	8.3	2.6
2500	710	9.7	10.7	8.2	2.4
2250	627	9.4	10.4	8.0	2.2
2000	547	9.1	10.1	7.9	2.0
1750	471	8.9	9.9	7.7	1.8
1500	396	8.6	9.6	7.5	1.6
1250	325	8.3	9.3	7.3	1.3
1000	255	8.1	9.1	7.1	1.1
750	188	7.8	8.8	6.8	0.9
500	124	7.5	8.5	6.6	0.6
250	61	7.3	8.3	6.2	0.3
100	24	7.1	8.1	6.0	0.1

T_{airhot}	T_{wgcold}	T_{wghot} - T_{airhot}		V_{wg}
[°C]	[°C]	[°C]		m^3/h
25	7.0	4.0		0.9

					Cooler
Q	V_{air}	T_{wghot}	$T_{aircold}$	LMTD	Surface
[W]	m^3	[°C]	[°C]	[°C]	Surface
5000	2353	12.4	16.4	10.9	3.6
4750	2169	12.1	16.1	10.9	3.4
4500	1994	11.8	15.8	10.9	3.3
4250	1830	11.6	15.6	10.8	3.1
4000	1675	11.3	15.3	10.8	2.9
3750	1528	11.0	15.0	10.7	2.8
3500	1389	10.8	14.8	10.7	2.6
3250	1257	10.5	14.5	10.6	2.4
3000	1131	10.2	14.2	10.6	2.2
2750	1012	9.9	13.9	10.5	2.1
2500	898	9.7	13.7	10.4	1.9
2250	790	9.4	13.4	10.3	1.7
2000	686	9.1	13.1	10.2	1.5
1750	587	8.9	12.9	10.2	1.4
1500	492	8.6	12.6	10.1	1.2
1250	402	8.3	12.3	9.9	1.0
1000	315	8.1	12.1	9.8	0.8
750	231	7.8	11.8	9.7	0.6
500	151	7.5	11.5	9.6	0.4
250	74	7.3	11.3	9.5	0.2
100	29	7.1	11.1	9.4	0.1

Airflow	[m^3]	600	900	1800	<3000	invalid

Table 12 Inlet air temperature 25°C

8.5 Change of coolant from water glycol/air density @2500m to pure water/air density @0m

Table 13 Coolant pure water, air density for sea level

9 ANALYSIS OF THE SCHROFF LX3 CABINET WITH INTEGRATED HEAT EXCHANGER

9.1 Radiated thermal power

The LX3 is a black cabinet. The power radiated by the surface of the cabinet is therefore approximately x1.4 higher than for the cabinet calculated in Table 5.

t[m]	0	0.01	0.02	0.032
k	5.50	2.10	1.30	0.89
ΔT	**Q_r**	**Q_r**	**Q_r**	**Q_r**
[°C]	**[W]**	**[W]**	**[W]**	**[W]**
0	0	0	0	0
5	190	72	45	31
10	379	145	89	61
15	569	217	134	92
20	758	290	179	123
25	948	362	224	153
30	1137	434	268	184
35	1327	507	313	215
40	1516	579	358	245

Table 14 Radiated power LX3 – Black → Q/0.7

9.2 Heat exchanger surface

The heat exchangers surface can be estimated by counting and measuring the fins. Each fin has a size of 175x19mm. The cabinet has 840 fins. This results in a cooler surface of about A_{LX3}= 2.8m^2.

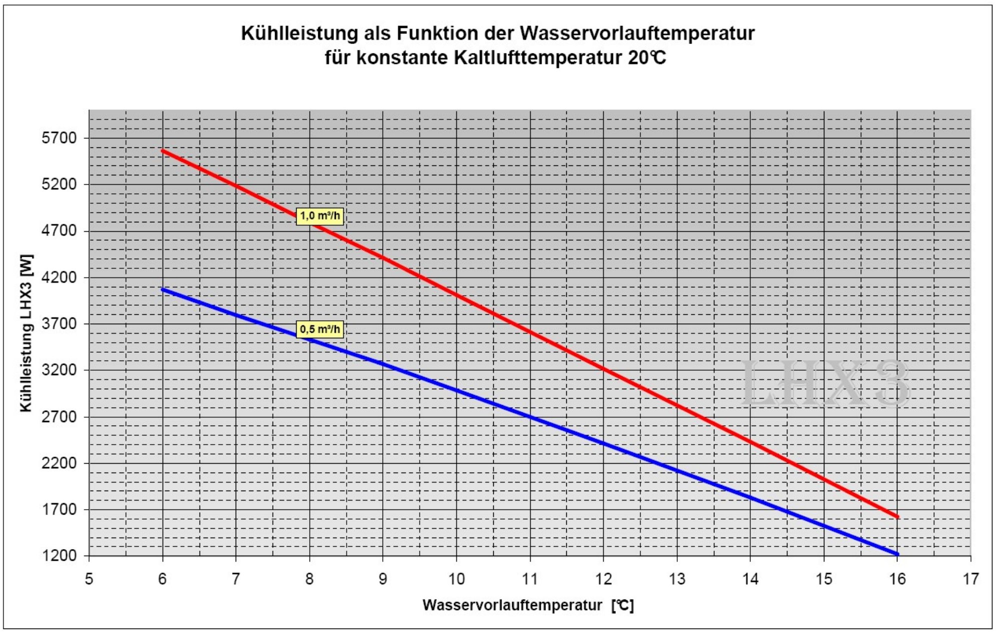


Figure 19 Cooling power as function of Twcold and Taircold = 20°C

The following parameters are given in the datasheet:

Airflow: 900m^3/h

Pressure loss water: 0.1bar @ Vw= 0.5m^3/h

Q = 3kW ; Twcool = 10°C, Taircool = 20°C, coolant water.

The following water/air temperatures are calculated from the graph above assuming the values shown below.

Air	
cpair	ρair
1004.8	1.225

Water	
cpwg	ρwg
4193.3	999.7026

Taircold	20	°C		Vair	900	m^3		A	2.8	m^2	

Q [W]	Vwater [m^3]	Twcold [°C]	Twhot [°C]	ΔTwater [°C]	Tairhot [°C]	ΔTair [°C]	Tairhot-Twatercold [°C]	Tairhot-Twaterhot [°C]	TairCold-Twaterhot [°C]	LMDT [°C]	FU [W/(K*m^2)]
5200	0.9	7	12.0	5.0	36.9	16.9	29.9	24.9	8.0	18.3	101.3
4000	0.9	10	13.8	3.8	33.0	13.0	23.0	19.2	6.2	14.1	101.3
3200	0.9	12	15.1	3.1	30.4	10.4	18.4	15.3	4.9	11.3	101.3
2025	0.9	15	16.9	1.9	26.6	6.6	11.6	9.6	3.1	7.1	102.3
3900	0.5	7	13.7	6.7	32.7	12.7	25.7	19.0	6.3	15.8	88.2
3000	0.5	10	15.2	5.2	29.7	9.7	19.7	14.6	4.8	12.2	88.2
2425	0.5	12	16.2	4.2	27.9	7.9	15.9	11.7	3.8	9.7	88.9
1525	0.5	15	17.6	2.6	25.0	5.0	10.0	7.3	2.4	6.1	89.4

Table 15

9.3 Airflow in cabinet

The airflow inside the cabinet depends on the operating point of the fans. The operating point of the fan can be read from the datasheet if the pressure loss in the system is known. The pressure loss depends on the air flow resistance in the cabinet. The airflow resistance of the empty LX3 is limited by the size of the air duct and the pressure drop across the heat exchanger. The equivalent hydraulic diameter of the air duct can be calculated by the formulae:

$d_h = 2*a*b/(a+b)$

$d_h = 2*0.28*0.039/(0.28+0.039) = 0.0684m$

The duct length is 1.853m.

The air in the cabinet flows on either side through a duct. Assuming an symmetric air flow (datasheet of 900m^3) results in 450m^3 in each air duct on either side.

The pressure loss can be then calculated using the equation shown in the earlier chapters.

Δp= 53.5 Pa; Tair = 20°C, altitude: 0m

Δp= 39.15 Pa; Tair = 20°C, altitude: 2500m

The duct is bend at the top and the bottom with an angle of about 30°.

From the chart in Figure 11 a pressure loss of about 30 Pa can be extrapolated per bend.

The total pressure loss per air duct is therefore:

Δptot = 53.5 Pa + 2x 30Pa = 113.5Pa

As the two ducts are in parallel, the pressure loss seen by the fan shall be assumed as Δptot/2.

The air pressure drop across the heat exchanger shall be neglected for simplicity as no data is available.

The fan fitted in the LX3 is a EBM Papst EBMPapst R2E220-AB06-05. The volume flow versus pressure loss is shown in the figure below (fan datasheet). The operating point for a pressure loss of 56.7 Pa is shown in the figure below.

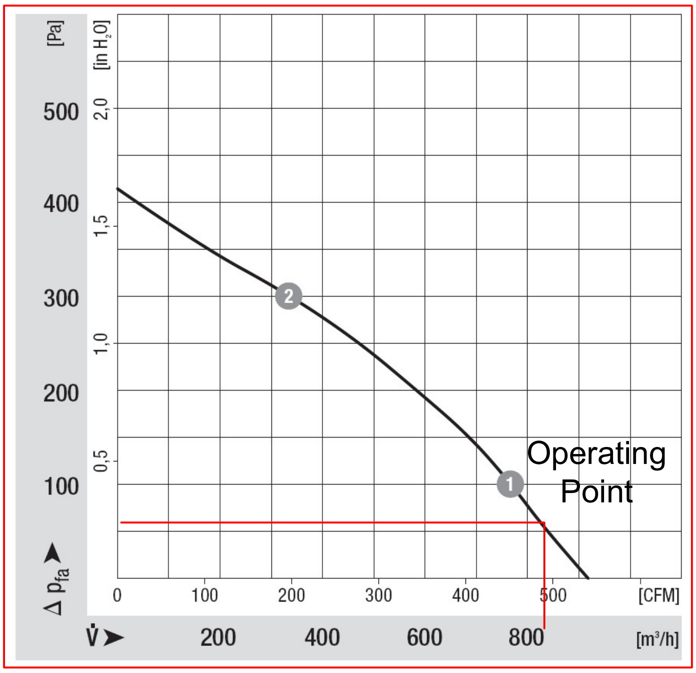


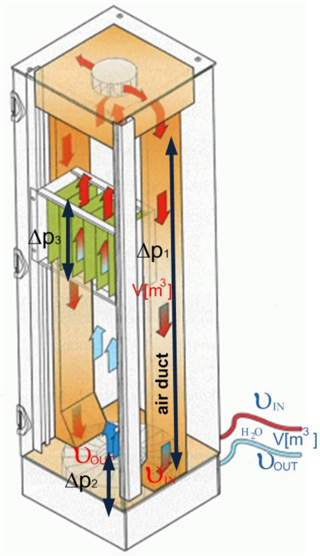
Figure 20 Fan characteristic: Pressure loss versus volume air flow (EBMPapst R2E220-AB06-05)

The fan delivers an airflow of around 830m^3.

9.4 Thermal cabinet load

A required thermal load of 1.5kW shall be assumed for the cabinet. From the Table 6 a ΔTwg = 1.6°C / 0.9m^3/h (2.9°C / 0.5m^3/h) and a ΔTair = 6.8°C can be found for a air flow of 900m^3.

The max coolant temperature is specified with 7°C for ambient 15°C. → The coolant Twgh=8.6°C. If the cabinet is insulated with 10mm thick armaflex foam, the max temperature inside the cabinet Tairhot < Tambient +10°C = 25°C to meet the thermal radiation specification of <150 W (see chapter 2). →Taircold = 25°C - 6.8°C = 18.2°C.



LMTD = [(25°C-8.6°C) – (18.2°C – 7°C)] / ln[(25°C-8.6°C)/(18.2°C – 7°C)] = 13.6 °C

Assuming a FU for the heat exchanger of <88 W / (K*m^2) as derived for the LX3 heat exchanger from Table 15, the required cooler surface can be calculated as follows:

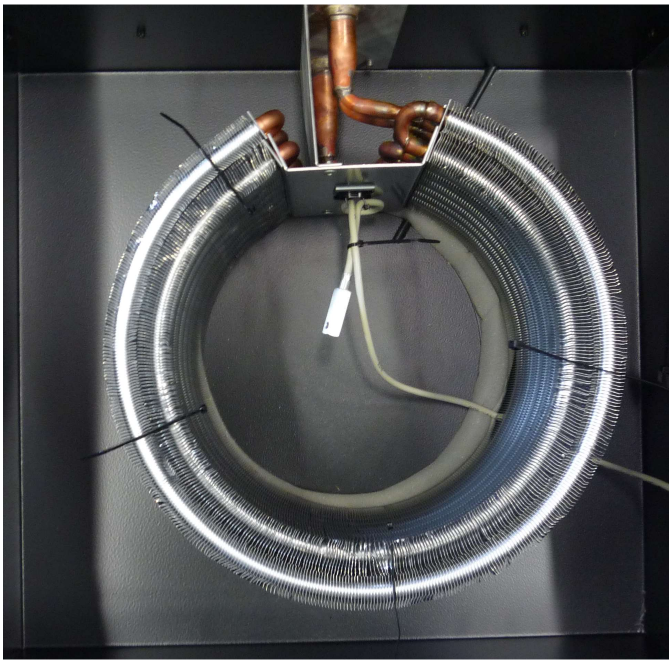
Areq = Q/(UF * LMT) = 1500W /(88 *13.6) ; W /(W*K/(K*m^2))

Areq = 1.25m^2 < A_{LX3}=2.8m^2

The cooler of the LX3 is therefore found to be sufficient for dissipation of 1.5kW under the conditions specified in section 2.

*Note: It should be mentioned that the UF= 88W/(K*m^2) is based on water as coolant. The water/glycol mixture as defined in section 2 is slightly less efficient →10%, but can be compensated through higher coolant flow rates.*

Use charts for LMDT



9.5 Air flow and air pressure loss in the cabinet

The air flow of the LX3 is specified with Vair 900m^3/h for the empty cabinet. Once racks are mounted inside the cabinet, the resistance to the airflow is increased. An air pressure drop occurs across the rack. The fan operating point changes according to the pressure drop → Volume flow of air decreases → cooling efficiency decreases. To compensate for the pressure drop, an additional fan must be placed in series to the rack as explained in section 5.5. The Knuerr Coolblast fan units provide 1U fan units including failure supervision with airflows of 495m^3/h to 1650m^3/h.

The fan units are equipped with 3x or 6x axial fans from ebmpapst type 4114N/2H.

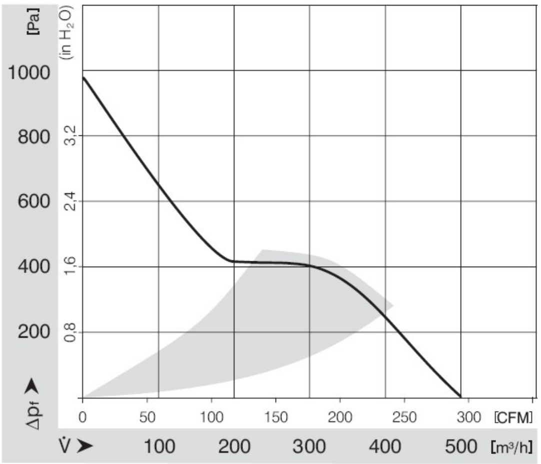


Figure 21 Fan 4114N/2H

A fan with higher volume flow will provide a better cooling for the rack mounted directly above or below the fan. Nevertheless, the overall volume flow through the heat exchanger is dependent on the volume flow through the heat exchanger. Therefore, the overall power extracted by the heat exchanger is in case of the LX3 cabinet limited by the fan serving the air duct supplying the airflow through the heat exchanger.

Note: Max thermal power extracted from a cabinet is limited by the volume air flow through the heat exchanger (assuming the volume flow of water/glycol is not the limiting factor). For cooling of particular electronics, fans trays with higher airflow rates might improve the cooling significantly in hot spots. Radial fans used in particular hotspots might provide better cooling than axial fans as the ΔP vs V is higher.

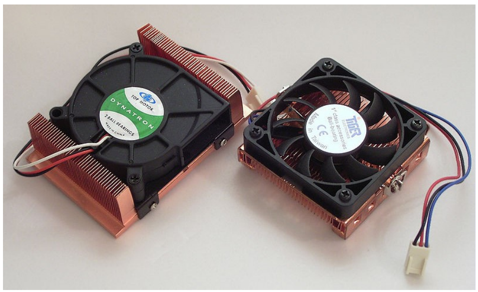


Figure 22 Fans with heat sink mounted for hot spots

10 SUMMARY

Various basics on thermal transfer has been discussed and calculated.

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