Cooling Load Calculations

1-Definitions:

- Space indicates either a volume or a site without a partition or a partitioned room or group of rooms.
- A room is an enclosed or partitioned space that is usually treated as a single load. Conditioned room often has an individual control system.
- A zone is a space , or several rooms, or units of space having some sort of coincident loads or similar operating characteristics. Zone may or may not be an enclosed space , or it may consist of many partitioned rooms. It could be a conditioned space or a space that is not air conditioned. A conditioned zone is always equipped with an individual control system.

2-Space and Equipment Loads:

The sensible and latent heat transfer between the space air and the surroundings can be classified as follows:

a) Space heat gain:

Space heat gain is the rate at which heat enters a conditioned space from an external source or is released to the space from an internal source during a given in time interval.

b) Space cooling load:

Space cooling load is the rate at which heat must be removed from a conditioned space so as to maintain a constant temperature and acceptable relative humidity.

c) Space heating load

Space heating load is the rate at which heat must be added to the conditioned space to maintain a constant temperature and sometimes a specified relative humidity.

d) Coil load:

- **Cooling coil load:** cooling coil load is the rate at which energy is removed at the cooling coil that serves one or more conditioned spaces equals the sum of the instantaneous space cooling loads
- **Heating coil load:** Heating coil load is the rate at which heat is added to the conditioned air from the hot water , steam , or electric heating elements inside the coil .

3- General Rules:

To calculate a space cooling load, detailed building design information and weather data at selected design conditions are required. Generally, the following steps should be followed:

a) Building characteristics. Building materials, components size, external surface colors, and shape are usually determined from building plans and specifications.

b) Determine building locations

c) Outdoor design conditions. Weather data may be obtained from local weather stations or from the National Climatic Center

d) Indoor design conditions. Such as indoor dry-bulb temperature, indoor wet bulb temperature, and ventilations rate. Include permissible

e) Operating schedules. Obtained a proposed schedule of lighting , occupants , internal equipment , applications , and processes that would contribute to the internal thermal load.

f) Date and time. Select the time of day and month to do the cooling load calculation

4- Heat Gain Classification According to Heat Source:

- **External heat sources:**

Source of external heat gains include the following:

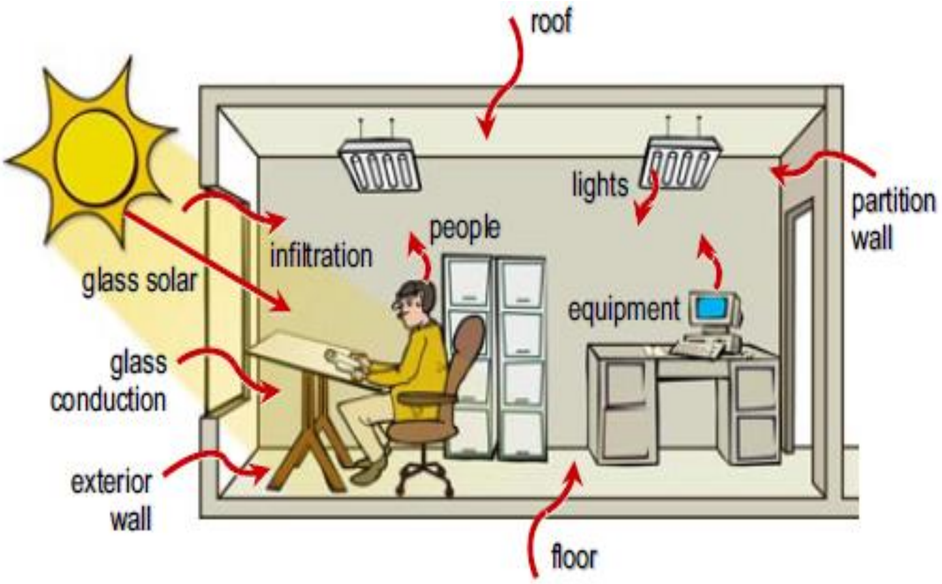
1- Heat gain entering from the exterior walls and roofs.
2- Solar heat gain transmitted through the glass
3- Heat gain entering from the partition walls and interior doors.
4- Infiltration of outdoor air into the conditioned space.

- **Internal heat sources:**

These heat gains are formed by the release of sensible and latent heat from the heat sources inside the conditioned space. These sources contribute internal heat gains as:

1- People.

2- Electric lights.



COOLING LOAD ESTIMATION

A- *External Heat Gain*

1-Heat gain from the walls

$Q_{wall} = A.\,U.\,CLTD_c$ → CLTD_c	Cooling Load Temperature Difference See Note (1)	°C	
→ U	Over all heat transfer coefficient for wall	W/ m² °C	**Table (6)**
→ A	Area of wall	m²	
→ Q_{wall}	Heat gain from walls	W	

Note 1: the values in Table (7) design on the

1. Outside dry blub temperature (35 ^{0}C)
2. Daily range (11.6)
3. inside dry bulb temperature (25.5 ^{0}C)

$$CLTD_C = (CLTD + LM)\,.\,K + (25.5 - T_i) + (T_{av} - 29.4)$$

CLTD: Cooling Load Temperature Difference from Table (7)
LM: Is latitude and month applied to walls from Table(9)
K: Correct factor of wall color
K=1 dark wall, K=0.83 medium color wall, K=0.65 light color
T_i : Inside Temperature (Room Temperature) ^{0}C
T_{av} = Outside Temperature – (Daily Range from Table (3))/2

2-Heat gain from roof

$Q_{roof} = A\,.\,U.\,CLTD_c$ → CLTD_c	Cooling Load Temperature Difference See Note (2)	°C	
→ U	Over all heat transfer coefficient for roof	W/ m² °C	**Table (5)**
→ A	Area of roof	m²	
→ Q_{roof}	Heat gain from roof	W	

Note 2: the values in Table (5) design on the

1. Outside dry blub temperature (35 ^{0}C)
2. Daily range (11.6)
3. inside dry bulb temperature (25.5 ^{0}C)

$$CLTD_C = [\,(CLTD + LM)\,.\,K + (25.5 - T_i) + (T_{av} - 29.4)]\,.\,f$$

CLTD: Cooling Load Temperature Difference from Table (5)
LM: Is latitude and month applied to walls from Table (9)
K: Correct factor of roof color

K=1 dark wall, K=0.83 medium color wall, K=0.65 light color

T_i: Inside Temperature (Room Temperature) ^0C

T_{av} = Outside Temperature – (Daily Range from Table (3))/2

f: Is a factor for attic fan and/or ducts above ceiling and is applied

f = 1 (no attic fan and duct)

f = 0.75 (attic fan and duct)

3-Solar radiation through glass

$$Q_{sol} = A \cdot SC \cdot SHGF \cdot CLF$$

CLF	Cooling Load Factor		**Table (13)**
SHGF	Solar Heat Gain	W/m^2	**Table (11)**
SC	Shading Coefficient		**Table (28)**
A	Area of net glasses	m^2	
Q_{sol}	Heat gain through glass	W	

4- conduction transmission gain through glass

$$Q_{tr} = A \cdot U \cdot CLTD_c$$

$CLTD_c$	Cooling Load Temperature Difference See Note (3)	^0C	
U	Over all heat transfer coefficient for glass	$W/m^2\,^oC$	**3.46**
A	Area of net glasses	m^2	
Q_{tr}	Heat gain from window	W	

Note 3: the values in Table (10) design on the

1. Outside dry blub temperature (35 ^0C)
2. Daily range (11.6)
3. inside dry bulb temperature (25.5 ^0C)

$$CLTD_C = CLTD + (25.5 - T_i) + (T_{av} - 29.4)$$

CLTD: Cooling Load Temperature Difference from **Table (10)**

T_i : inside temperature (room temperature) ^0C

T_{av} = outside temperature – (daily range)/2

5-Heat transmission Partition

$$Q_{part.} = A \cdot U \cdot (T_b - T_i)$$

$(T_b - T_i)$	Temp. of Room No Air- condition & Temp. of Room Air- Condition	^oC	
U	Over all heat transfer coefficient for partition	$W/m^2\,^oC$	**Table (6)**
A	Area of partition between two room	m^2	
$Q_{part.}$	Heat gain from partition	W	

$$T_b = T_i + 2/3\,(T_o - T_i)$$

T_i : Room Temperature or (Inside Temperature) T_o : Outside Temperature

B- *Internal Heat Gain*

6-Heat gain from people

a-Sensible heat gain

Q_s= No . (Sensible Heat Gain) . CLF	Cooling Load Factor		
	Sensible heat gain for people	W/person	**Table 18**
	No. of people in space		
	Heat gain from people	W	

b-Latent heat gain

Q_L= No . (Latent Heat Gain)	Latent heat gain for people	W/person	**Table 18**
	No. of people in space		
	Heat gain from people	W	

7-Heat gain from light

Qs = A . q . CLF	Cooling Load Factor		
	Lighting intensity	W/ m^2	**Table 19**
	Area of floor		
	Heat gain from light	W	

8-Heat gain from appliances and motors

a-Sensible heat gain from appliances

Q_s= No . (Sensible Heat Gain) . CLF	Cooling Load Factor		**Table(22)&(23)**
	Sensible heat gain from appliances	W/app	**Table (21)**
	No. of appliances		
	Heat gain from appliances	W	

b-Latent heat gain from appliances

Q_L= No . (Latent Heat Gain)	Latent heat gain from appliances	W/app	**Table (21)**
	No. of appliances		
	Heat gain from appliances	W	

c-Sensible heat gain from motors

$Q_{s/motor} = P$ →	Heat gain from motor	W	

9-Heat gain due to infiltration

a- Sensible heat gain OASH

$Q_s = 1.232 \cdot V \cdot (T_o - T_i)$	Outdoor & Indoor temperatures	ºC	
	Volume flow rate of air See **Note 4**	*l/s*	
	Heat lost due Ventilation	W	

Note 4:
There are three methods commonly used to estimate infiltration

1. Crack length method
2. Air change method
3. Effective leakage – area method

The air change method is easiest

$\dot{V} = \frac{H.W.L.n.10^3}{3600}$	Room Height, width and length	m^3	
	n: Air change per hour		Table (3.18)
	Volume flow rate of air	*l/s*	

b- Latent heat gain OALH

$Q = 2940.\dot{V}(w_o - w_i)$	Outdoor & Indoor moisture content	kg/kg_{air}	
	Volume flow rate of air See Note 4	*l/s*	
	Heat lost due Ventilation	W	

10-Heat gain due to ventilation

a- Sensible heat gain OASH

$Q_s = 1.22 \cdot V \cdot (T_o - T_i)$	Outdoor & Indoor temperatures	ºC	
	Volume flow rate of air See **Note 5**	*l/s*	
	Heat lost due Ventilation	W	

Note 5:

There are three methods commonly used to estimate ventilation

1. Number of occupants (l/s / person) method
2. Volume of the space (m^3air / m^3space) method
3. Floor area of the space (m^3/ square meter of floor) method

The Number of occupants (l/s / person) method is easiest

$\dot{V} = No.v$	Air required per person	*l/s.per.*	Table (2.7)
	No. of persons		
	Volume flow rate of air	*l/s*	

b- Latent heat gain OALH

$Q_{vent.} = 2940.\dot{V}(w_o - w_i)$	Outdoor & Indoor moisture content	kg/kg $_{air}$	
	Volume flow rate of air See **Note 5**	*l/s*	
	Heat lost due Ventilation	W	

ROOM LOAD

11-**R**oom **S**ensible **H**eat ***RSH***

- $RSH = \sum equs.(1,2,3,4,5,6a,7,8a,8c)$

12-**R**oom **L**atent **H**eat ***RLH***

- $RLH = \sum equs.(6b + 8b)$

13-**R**oom **T**otal **H**eat ***RTH***

- $RTH = RSH + RLH$

14-**T**otal **S**ensible **H**eat ***TSH***

- $TSH = RSH + Either \quad equ.\, 9a \,\&\, 10a$

15-**T**otal **L**atent **H**eat ***T l H***

- $TLH = RLH + Either \quad equ.\, 9b \,\&\, 10b$

16-**G**rand **T**otal **H**eat ***GTH***

- $GTH = TSH + TLH$

$$COOLING\ LOAD \quad in \quad (Ton\ refrigeration) = \frac{GTH\ (in \quad watt)}{3516.3}$$

AIR DUCT DESIGN

Introduction

Commercial, industrial, and residential air duct system design must consider (ASHRAE, 2001)

1. Space availability
2. Space air diffusion
3. Noise levels
4. Duct leakage
5. Duct heat gains and losses
6. Balancing
7. Initial investment cost
8. System operating cost

General Rules For Duct Design

1. Air should be conveyed as directly as possible to save space, power and material
2. Sudden changes in directions should be avoided. When not possible to avoid sudden changes, turning vanes should be used to reduce pressure loss
3. Diverging sections should be gradual. Angle of divergence ≤ 20
4. Aspect ratio should be as close to 1.0 as possible. Normally, it should not exceed 4
5. Air velocities should be within permissible limits to reduce noise and vibration

Air Duct System Components

Air ducts can be classified into four types according to their transporting functions:

1. ***Supply duct***: Conditioned air is supplied to the conditioned space
2. ***Return duct***: Space air is returned to the fan room where the air handling unit is installed and the packaged unit
3. ***Outdoor air duct*** : outdoor air is transported to the air handling unit, to the fan room, or to the space directly

4. ***Exhaust duct*** : space air or contaminated air is exhausted from the space, equipment, and fan room

Duct Sizing Methods

Duct sizing determines the dimensions of each duct section in the air duct system calculating the total pressure loss of the air duct system. Four duct-sizing methods are currently used:

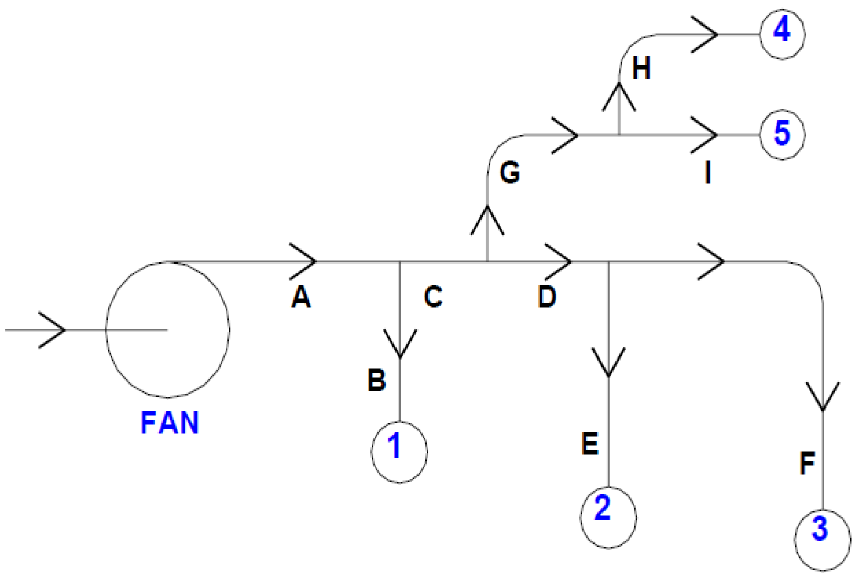
1. Equal-friction method with maximum velocity.
2. Static regains method.
3. Constant velocity method.
4. T method

Equal-frication method with maximum velocity is the most common method for duct design.

Equal-Friction Method With Maximum Velocity

In this method the frictional pressure drop per unit length in the main and branch ducts (Δp/L) are kept same, i.e.,

$$\left(\frac{\Delta p_f}{L}\right)_A = \left(\frac{\Delta p_f}{L}\right)_B = \left(\frac{\Delta p_f}{L}\right)_C = \left(\frac{\Delta p_f}{L}\right)_D = \quad \ldots\ldots\ldots\ldots \quad (1)$$



Then the stepwise procedure for designing the duct system is as follows:

1. Select a suitable frictional pressure drop per unit length (Δp/L) so that the combined initial and running costs are minimized.
2. Then the equivalent diameter of the main duct (A) is obtained from the selected value of (Δp/L) and the airflow rate. As shown in Fig.(1), airflow rate in the main duct Q_A is equal to the sum total of airflow rates to all the conditioned zones, i.e.,

$$\dot{Q}_A = \dot{Q}_1 + \dot{Q}_2 + \dot{Q}_3 + \dot{Q}_4 + \dot{Q}_5 = \sum_{i=1}^{N} \dot{Q}_i \qquad \text{.......... (2)}$$

From the airflow rate and (Δp/L) the equivalent diameter of the main duct ($D_{eq,A}$) can be obtained either from the friction chart or using the frictional pressure drop equation, i.e.,

$$D_{eq,A} = \left(\frac{0.022243\,\dot{Q}_A^{\,1.852}}{\left(\frac{\Delta p_f}{L}\right)_A} \right)^{\left(1/4.973\right)} \qquad \text{.......... (3)}$$

3. Since the frictional pressure drop per unit length is same for all the duct runs, the equivalent diameters of the other duct runs, B to I are obtained from the equation:

$$\left(\frac{\dot{Q}^{1.852}}{D_{eq}^{\,4.973}} \right)_A = \left(\frac{\dot{Q}^{1.852}}{D_{eq}^{\,4.973}} \right)_B = \left(\frac{\dot{Q}^{1.852}}{D_{eq}^{\,4.973}} \right)_C = \qquad \text{.......... (4)}$$

4. If the ducts are rectangular, then the two sides of the rectangular duct of each run are obtained from the equivalent diameter of that run and by fixing aspect ratio as explained earlier. Thus the dimensions of the all the duct runs can be obtained. The velocity of air through each duct is obtained from the volumetric flow rate and the cross-sectional area.

5. Next from the dimensions of the ducts in each run, the total frictional pressure drop of that run is obtained by multiplying the frictional pressure drop per unit length and the length, i.e.,

$$\Delta P_{f,A} = \left(\frac{\Delta p_f}{L}\right)_A . L_A \; ; \; \Delta P_{f,B} = \left(\frac{\Delta p_f}{L}\right)_B . L_B \qquad \text{..........} \; \textbf{(5)}$$

6. Next the dynamic pressure losses in each duct run are obtained based on the type of bends or fittings used in that run.

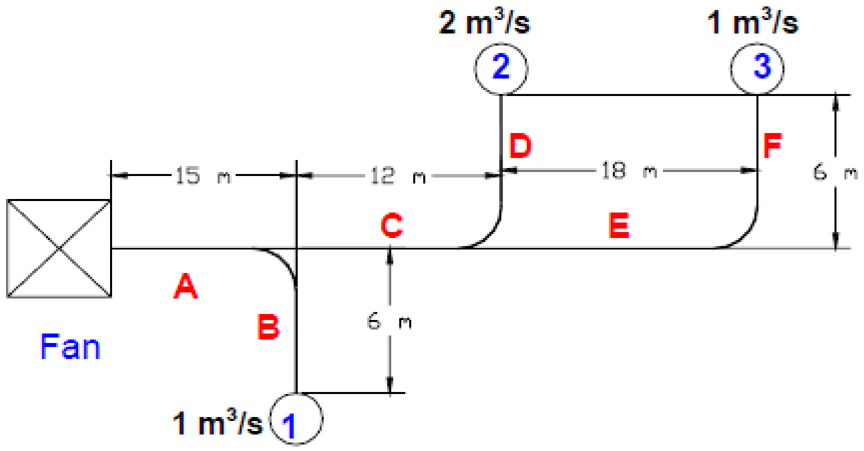
7. Next the total pressure drop in each duct run is obtained by summing up the frictional and dynamic losses of that run, i.e.,

$$\Delta P_A = \Delta p_{f,A} + \Delta p_{d,A} \; ; \; \Delta P_B = \Delta p_{f,B} + \Delta p_{d,B} \qquad \text{..........} \; \textbf{(6)}$$

8. Next the fan is selected to suit the index run with the highest pressure loss. Dampers are installed in all the duct runs to balance the total pressure loss.

Equal friction method is simple and is most widely used conventional method. This method usually yields a better design than the velocity method as most of the available pressure drop is dissipated as friction in the duct runs, rather than in the balancing dampers.

Example: - The following figure shows a typical duct layout. Design the duct system using Equal friction method. Take the velocity of air in the main duct (A) as 8 m/s for the methods. Assume a dynamic loss coefficient of 0.3 for upstream to downstream and 0.8 for upstream to branch and for the elbow. The dynamic loss coefficients for the outlets may be taken as 1.0. Find the FTP required for each case and the amount of damping required.



Ans.:

$Q_A = 4\ m^3/s$, $V_A = 8\ m/s$

$A_A = Q_A / V_A = 4/8 = 0.5\ m^2$

$D_A = 0.798m$

$\Delta P_A / L_A = (0.022243 Q_A^{1.852}) \ / \ (D^{4.973})$

$\mathbf{\Delta P_A / L_A = 0.879\ pa/m}$

$$D_{e\ B} = \left(\frac{0.022243\, \dot{Q}_A^{\ 1.852}}{\left(\frac{\Delta p_f}{L}\right)_A} \right)^{\left(1/4.973\right)}$$

D_B= 0.4762 m , V_B= 5.6 m/s

D_C= 0.717 m, V_C= 7.5 m/s

D_D= 0.6164 m, V_B= 6.6 m/s

$D_{E,F}$= 0.4762 m, $V_{E,F}$= 5.6 m/s

Calculation of total pressure drop:

From fan to 1:

$\Delta P_{A-B} = \Delta P_{A,f} + \Delta P_{B,f} + \Delta P_{u-b} + \Delta P_{exit}$

$\Delta P_{u-b}=C_{u-b} * (\rho V^2) / 2$, $\Delta P_{exit}= C_{exit} * (\rho V^2) / 2$

$\Delta P_{A-B} = 13.35 + 5.34 + 15.1 + 18.9 =$ 52.69 Pa

From fan to 2:

$\Delta P_{A-C-D} = \Delta P_{A,f} + \Delta P_{C,f} + \Delta P_{D,f} + \Delta P_{u-d,C} + \Delta P_{u-b,d} + \Delta P_{exit}$

$\Delta P_{A-C-D} = 13.35 + 10.68 + 5.34 + 9.94 + 21.55 + 26.9 =$ 87.76 Pa

From fan to exit 3:

$\Delta P_{A-C-E-F} = \Delta P_{A,f} + \Delta P_{C,f} + \Delta P_{E,f} + \Delta P_{F,f} + \Delta P_{u-d,C} + \Delta P_{u-d,E} + \Delta P_{elbow} + \Delta P_{exit}$

$\Delta P_{A-C-E-F} = 13.35 + 10.68 + 16.02 + 5.34 + 9.94 + 5.67 + 15.1 + 18.9 =$95pa

The required FTP is:

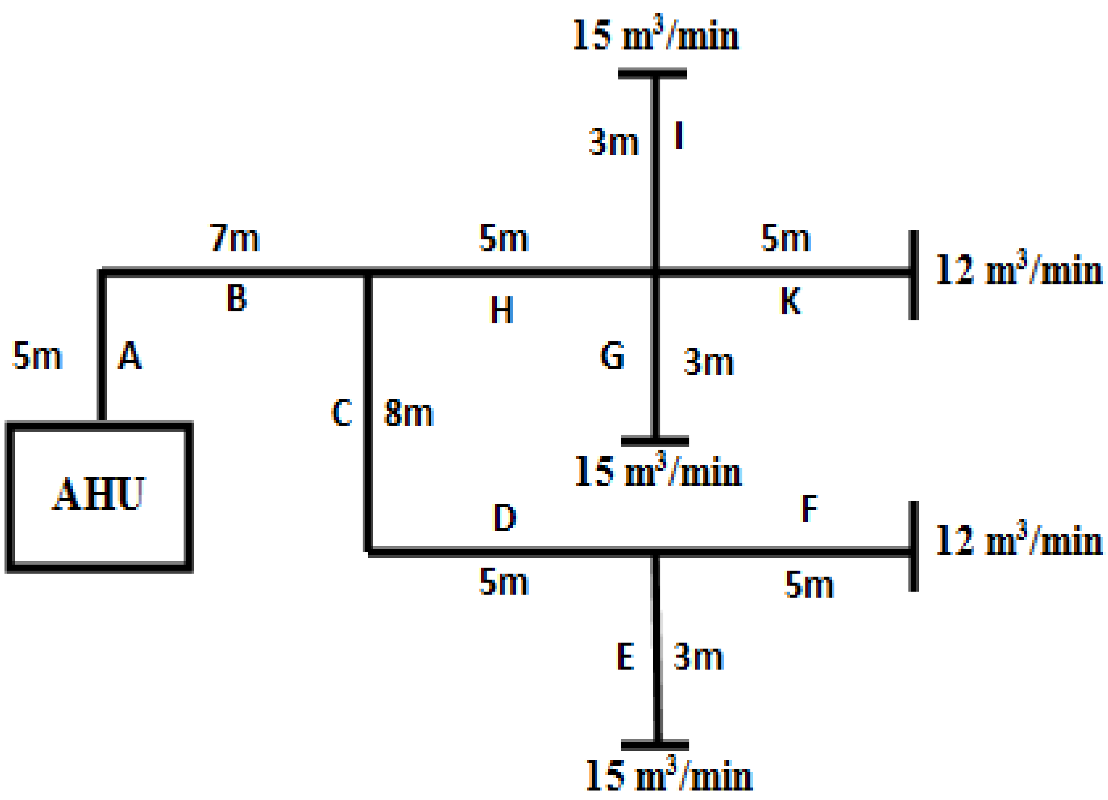
FTP= $\Delta P_{A-C-E-F}$= **95pa**

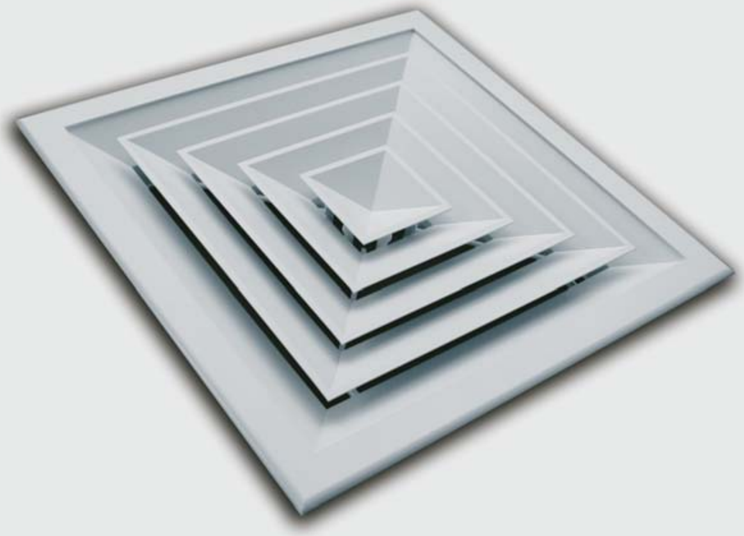
Amount of dampering required at 1 = FTP - ΔP_{A-B} = **42.31 Pa**

Amount of dampering required at 2 = FTP - ΔP_{A-C-D} = **7.3 Pa**

H.W :- Find the duct sizes and fan total pressure (F.T.P) by using equal frication method for the duct system show in figure. Take the dynamic loss coefficient as follows: -

1. 0.3 for upstream to downstream
2. 0.8 for upstream to branch
3. 0.8 for 90˚ elbow
4. 1 for outlet





CDA

Louvred Face **Ceiling Diffuser**

KES KLiMA
AIR CONTROL SYSTEMS

Properties

The CDA type diffusers have fixed and straight blades. For supply air purposes, they are characteristically suitable for horizontal air throws. Where "Coanda effect" is required, they should be installed close to the ceiling. These diffusers are recommended for use with ceiling heights up to 4 m., with a supply air temperature difference of (+/-) 10°C.The diffuser is made of a frame and a central blade block. The blade block is fixed to the frame by the aid of spring pins and can easly be removed / installed. The standard sizes start from 150 x 150 mm, and go up to 600 x 600 mm with increments of 75 mm. One, two, three and four way throwing types are available.

Materials

The frame and the blades are manufactured from ETIAL-60 norm aluminium profiles.

Accessories

Damper With Opposed Blades

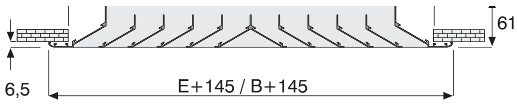
Depending on application characteristics, an opposed blade damper can be installed on the back side of the diffuser. This damper is a separate item which can be operated by its special tool from the face of the diffuser. Opposed blade dampers are manufactured from ETIAL-60 norm aluminium extruded profiles. To prevent reflection, they are painted RAL 9005 (matt black) as standard.

Flap Damper With Rectifier

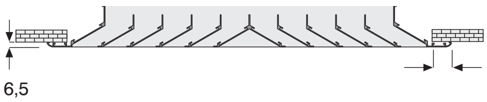
This type of damper is used in high velocity ducts. The rectifier is made of ETIAL-60 norm aluminium profiles and the flap damper part is formed from steel sheets. To prevent reflection, they are painted RAL 9005 (matt black) as standard

Plenum Box

The plenum box is used to achieve optimum throw characteristics. It has the inlet either at the top or at one side. Depending on request, a damper can be installed at the inlet, which can be operated internally or externally (has to be specified with the order).The plenum boxes are made from 0.6 mm thick galvanized steel sheets and have 4 hanging brackets on their body.Optionally, a 6 mm thick acoustic foam can be laid inside the plenum box.



CDA.32.AA



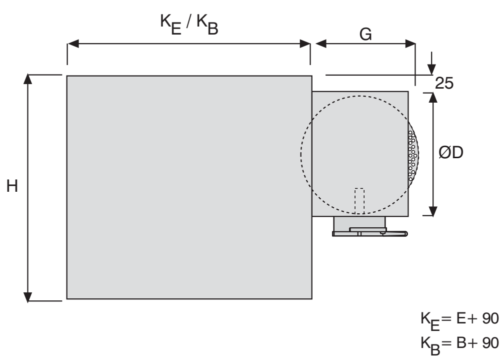
CDA.32.AA

Standard Dimensions (mm)

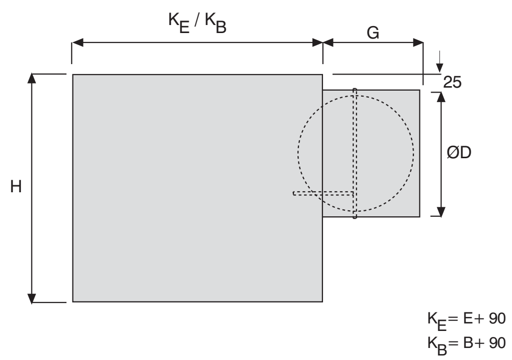
E	B	ØD	H	G
150	150	170	250	150
	225	244	350	175
	300	244	350	175
	375	295	450	225
	450	346	450	225
	525	346	450	225
	600	346	450	225
225	225	244	350	175
	300	295	400	225
	375	346	450	225
	450	396	500	250
	525	396	500	250
	600	447	550	275

E	B	ØD	H	G
300	300	295	450	225
	375	346	450	225
	450	447	550	275
	525	447	550	275
	600	498	600	300
375	375	396	500	250
	450	447	550	275
	525	498	600	300
	600	498	600	300
450	450	498	600	300
	525	498	600	300
	600	498	600	300
525	525	498	600	300
	600	498	600	300
600	600	498	600	300

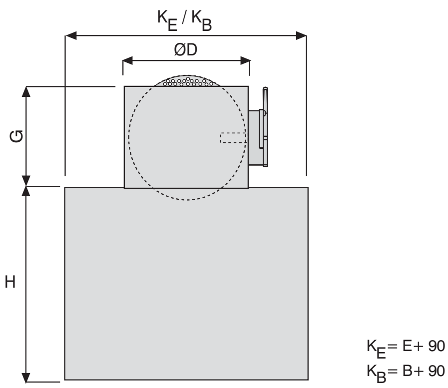
Externally Operated Side Inlet



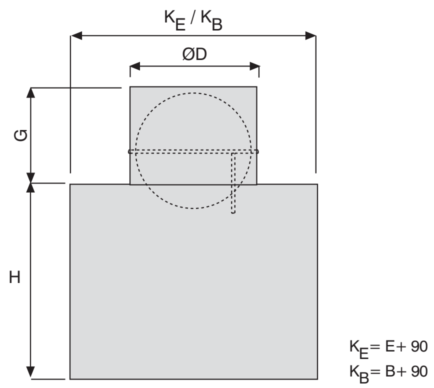
Internally Operated Side Inlet



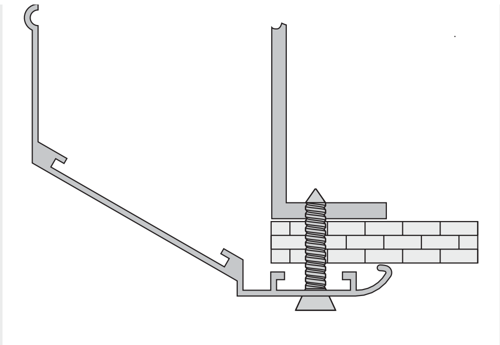
Externally Operated Top Inlet



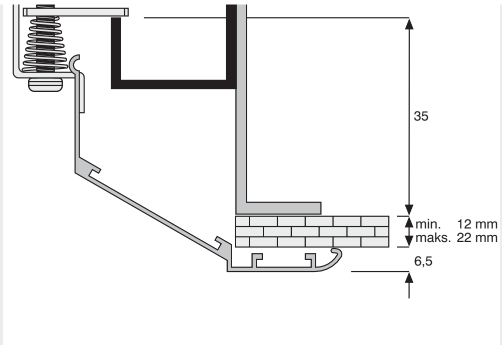
Internally Operated Top Inlet



KES KLIMA

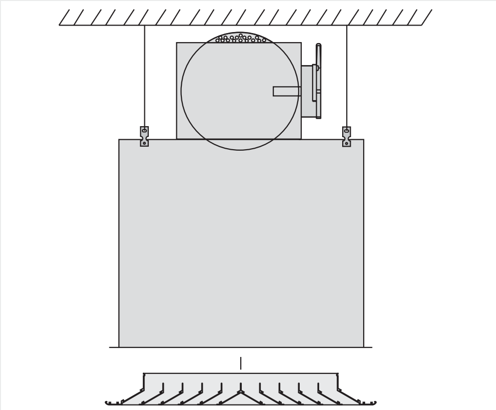


A set of Ø 4.2 x 38 mm self-drilling screws, painted the same, are given with the product



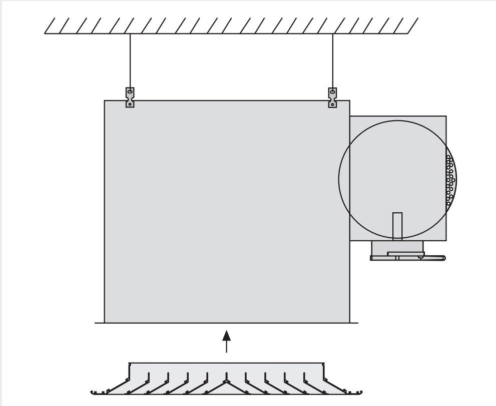
Suitable for ceiling thickness 12-22 mm. For other thicknesses, please contact us.

Plenum box installation (top inlet)



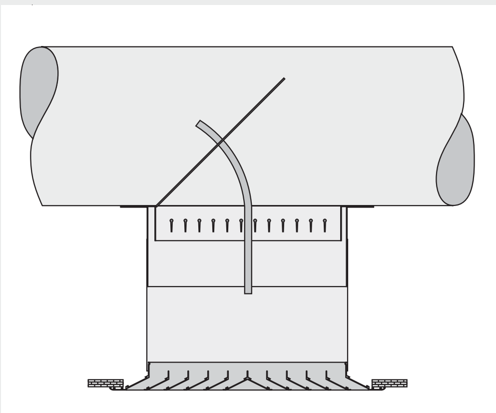
There are 4 hanging brackets on the box as standard

Plenum Box installation (side inlet)

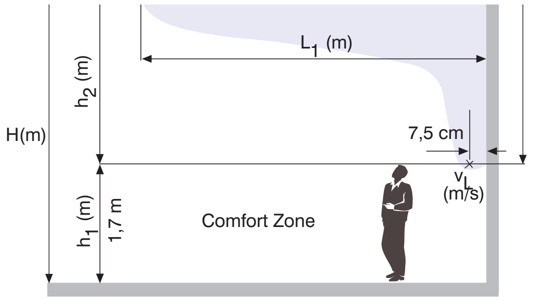


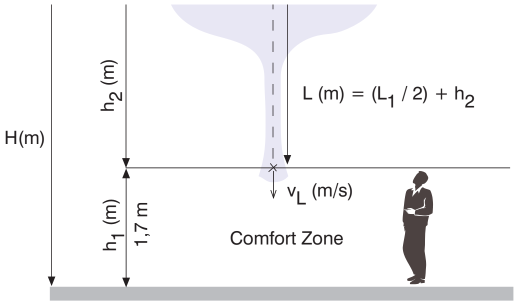
There are 4 hanging brackets on the box as standard

Application of flap damper with rectifier



Applied in high speed ducts





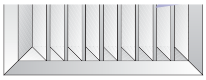
L_1	Distance between diffuser centres or diffuser centre and wall. (m)
h1	Comfort zone height (m)
h2	Distance between a diffuser and comfort zone (m)
v_{eff}	Effective outlet velocity (m/s)
v_L	Velocity of core in comfort zone
Δt_0	Difference between supply air and room temperature (^oC)
Δt_L	Difference between core and comfort zone temperature (^oC)
L	Throw distance (m)
V	Air flow rate (m3/h)
H	Room height (m)
S	Sound power level dB(A)

To achieve "Coanda effect", the outlet velocity must be greater than 2m/s. The general comfort conditions require that the sound power level is below 40 dB(A). The height of the comfort zone is taken as 1.70m above the floor. It is important that 0.25 m/s core velocity is not exceeded in this zone.

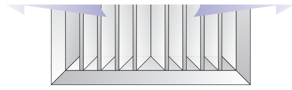
Note: The tables are given for 4 types of blade blocks (11,21,24,41). For other types of blocks listed on page 11, please contact us.

	Sound power level	pressure drop
Supply air, with damper	+3 dB (A)	x 1,0
Extract air	+3 dB (A)	x 1,1
Extract air with damper	+13 dB (A)	x 1,15

The data given in the tables are valid for supply air, without dampers. For other conditions, the correction factors in the table (left) have to be applied



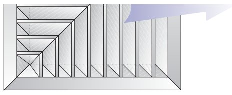
Size E/B (mm)	Flow Rate V(m^3/h)	Throw, L (m)		Pressure loss ΔP (Pa)	Sound power level S (dB(A))
		v_L=0,25 m/s	v_L=0,10 m/s		
150 x 150	120	1,50	2,50	9	<20
	160	2,00	3,50	15	<20
	200	2,20	4,50	23	25
	250	2,50	5,50	33	29
	280	3,00	6,00	43	33
225 x 225	280	2,00	4,00	9	<20
	370	2,50	5,50	15	25
	460	3,50	6,50	23	30
	550	4,00	8,00	33	34
	640	4,50	9,00	43	37
300 x 300	490	2,50	5,50	9	<20
	650	3,50	6,50	15	28
	810	4,50	8,50	23	33
	970	5,00	10,50	33	37
	1130	6,50	12,00	43	40
375 x 375	760	3,50	6,50	9	<20
	1010	4,50	8,50	15	30
	1270	5,50	11,00	23	35
	1520	6,50	13,50	33	39
	1770	7,50	15,00	43	42
450 x 450	1100	3,50	7,00	9	<20
	1460	5,50	10,00	15	30
	1820	6,50	13,00	23	40
	2190	7,50	16,00	33	40
	2550	9,00	18,00	43	45
525 x 525	1490	4,00	8,00	9	25
	1980	5,50	10,00	15	35
	2480	6,50	13,00	23	40
	2980	7,00	14,00	33	45
	3470	9,50	18,50	43	45
600 x 600	1950	4,50	8,50	9	25
	2590	6,00	11,00	15	35
	3240	7,00	14,00	23	40
	3890	8,50	17,00	33	45
	4540	10,00	19,00	43	50



Size E/B (mm)	Flow Rate V(m^3/h)	Throw, L (m)		Pressure loss ΔP (Pa)	Sound power level S (dB(A))
		v_L=0,25 m/s	v_L=0,10 m/s		
150 x 150	120	1,00	2,00	9	<20
	160	1,20	2,50	15	<20
	200	1,50	3,00	23	25
	250	2,00	3,50	33	29
	280	2,20	4,50	43	32
225 x 225	280	1,50	2,50	9	<20
	370	2,00	3,50	15	25
	460	2,50	5,00	23	30
	550	2,70	5,50	33	34
	640	3,00	6,00	43	37
300 x 300	490	2,00	3,50	9	<20
	650	2,50	5,00	15	28
	810	3,00	6,00	23	32
	970	3,50	7,50	33	37
	1130	4,50	8,50	43	40
375 x 375	760	2,50	4,50	9	<20
	1010	3,00	6,00	15	30
	1270	4,00	7,50	23	35
	1520	5,00	9,50	33	39
	1770	5,50	11,00	43	42
450 x 450	1100	2,50	6,00	9	26
	1460	3,50	7,50	15	32
	1820	4,50	9,00	23	37
	2190	5,50	11,00	33	40
	2550	6,50	13,00	43	44
525 x 525	1490	3,00	6,50	9	30
	1980	4,00	8,00	15	35
	2480	5,00	10,00	23	40
	2980	6,50	13,00	33	45
	3470	7,50	15,00	43	45
600 x 600	1950	3,50	7,00	9	30
	2590	4,50	8,50	15	35
	3240	5,50	10,50	23	40
	3890	6,50	13,00	33	45
	4540	7,00	15,00	43	45

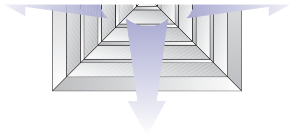
Refrigeration & AC /2 *Third Year* *Dr. Ayad Suleiman Abdullah*

2025 – 2026

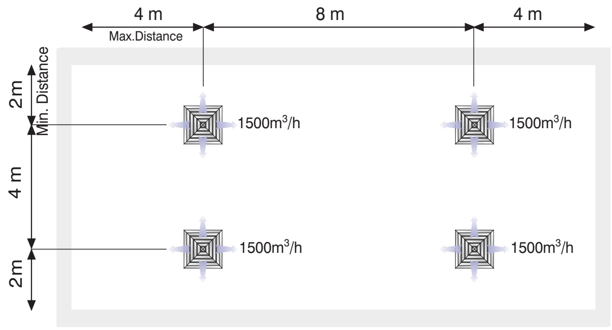


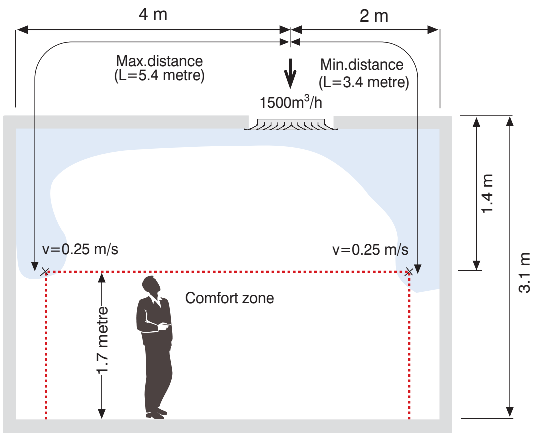
Size E/B (mm)	Flow Rate V(m^3/h)	Throw, L (m)		Pressure loss ΔP (Pa)	Sound power level S (dB(A))
		v_L=0,25 m/s	v_L=0,10 m/s		
150 x 150	120	1,00	2,00	9	<20
	160	1,10	2,50	15	<20
	200	1,50	3,00	23	20
	250	2,00	3,50	33	24
	280	2,00	4,50	43	28
225 x 225	280	1,50	2,50	9	<20
	370	2,00	3,50	15	20
	460	2,50	5,00	23	25
	550	2,70	5,50	33	29
	640	3,00	6,00	43	32
300 x 300	490	2,00	3,50	9	<20
	650	2,50	5,00	15	23
	810	3,00	6,00	23	28
	970	3,50	7,50	33	32
	1130	4,50	8,50	43	35
375 x 375	760	2,50	4,50	9	<20
	1010	3,00	6,00	15	25
	1270	4,00	7,50	23	30
	1520	5,00	9,50	33	34
	1770	5,50	11,00	43	37
450 x 450	1100	2,50	6,00	9	21
	1460	3,50	7,50	15	27
	1820	4,50	9,00	23	32
	2190	5,50	11,00	33	35
	2550	6,50	13,00	43	39
525 x 525	1490	3,00	6,50	9	25
	1980	4,00	8,00	15	30
	2480	5,00	10,00	23	35
	2980	6,00	12,00	33	40
	3470	7,00	14,00	43	40
600 x 600	1950	3,00	7,00	9	25
	2590	4,00	9,00	15	30
	3240	5,00	11,00	23	35
	3890	6,00	13,00	33	40
	4540	7,00	15,00	43	40

KES KLIMA



Size E/B (mm)	Flow Rate V(m^3/h)	Throw, L (m)		Pressure loss ΔP (Pa)	Sound power level S (dB(A))
		v_L=0,25 m/s	v_L=0,10 m/s		
150 x 150	120	1,00	1,50	9	<20
	160	1,10	2,00	15	<20
	200	1,50	2,50	23	<20
	250	1,70	3,00	33	<20
	280	2,00	3,50	43	19
225 x 225	280	1,00	2,00	9	<20
	370	1,50	2,50	15	<20
	460	2,00	3,50	23	<20
	550	2,10	4,00	33	23
	640	2,20	4,50	43	27
300 x 300	490	1,50	2,50	9	<20
	650	2,00	3,50	15	20
	810	2,20	4,50	23	25
	970	2,50	5,00	33	29
	1130	3,00	6,00	43	32
375 x 375	760	2,00	3,50	9	<20
	1010	2,20	4,50	15	24
	1270	2,50	5,50	23	29
	1520	3,50	6,50	33	33
	1770	4,00	7,50	43	37
450 x 450	1100	2,00	4,00	9	22
	1460	2,50	5,00	15	28
	1820	3,50	6,50	23	33
	2190	4,00	8,00	33	36
	2550	4,50	9,00	43	40
525 x 525	1490	2,50	5,00	9	25
	1980	3,00	6,00	15	30
	2480	4,00	8,00	23	35
	2980	4,50	9,00	33	40
	3470	5,00	10,50	43	45
600 x 600	1950	2,50	5,50	9	30
	2590	3,50	7,00	15	35
	3240	4,50	8,50	23	40
	3890	5,00	10,50	33	45
	4540	6,00	12,00	43	45





Solution:

1) Diffusers are placed on the ceiling plan symmetrically.

2) Air flow rate per diffuser is calculated as 6000 / 4 = 1500 m3/h.

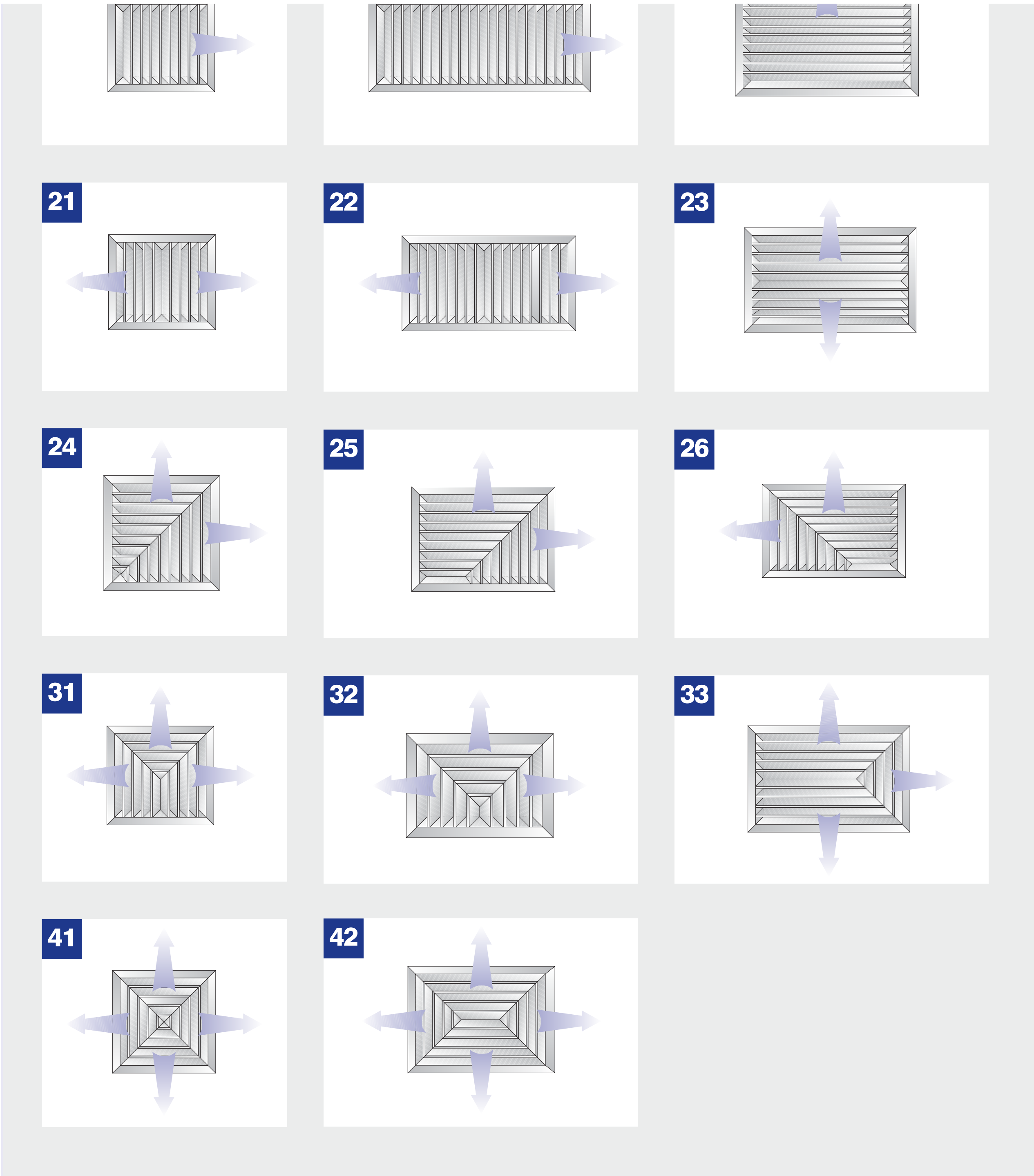
3) Calculation of path length to the comfort zone:

Minimum distance: L= 2.0 + 1.40 = 3.40 m

Maximum distance: L= 4.0 + 1.40 = 5.40 m.

4) From the table on page 9, the most suitable size is found as 375x375 mm; for 1500 m3/h and 3.40 m throw.

5) From the same table with interpolation, pressure loss is read as 32 Pa and sound power level as 33 dB(A).



Note: The views shown are face views, and throw directions are as seen from below. For blocks 25 and 26 care should be taken when ordering.

a central blade block. The blade block will be fixed to the frame by the aid of spring pins and will be easy to be removed / installed.Optionally, a damper will be installed on the back side of the diffuser. This damper will be a separate item which will be formed from ETIAL-60 norm aluminium profiles and be operated from the face of the diffuser. To prevent reflection, the damper will be painted RAL 9005 (matt black).The plenum box will be manufactured from

Class 0) will be installed inside the plenum box.

Order Code

Model CDA.32.AA.1 1-375 x 375 - 41 - 9010

Frame	32 mm
Accessories	**AA**..Without accessories **ZA**..Opposed blade damper
Installation	**0**.....Without screw holes **1**.....With screw holes **3**.....Concealed fixing
Installation accessories	**0**.....Without installlation bridge **1**.....With installation bridge
Standard Dimensions	E x B (mm) Refer to page 3
block code	Refer to page 11 11, 12, 13, 21, 22, 23, 24, 25, 26, 31, 32, 33, 41, 42
Colour Code	indicate RAL colour code

Plenum Box Order Code

Model PLA.10.S B.1 1-465 x 465 x 500 x 396 x 1

Installation	**10**...With Screws **30**...Concealed Fixing
Box Inlet	**S** ....Side Inlet **T**.....Top Inlet
spigot Damper	**A** ....Without Damper **B** ....Externally Operated **C** ....Internally Operated
Perforated Rectifier Plate	**0**.....Without Plate **1**.....With Plate
Insulation	**0**.....Without Insulation **1**.....With Acoustic Insulation
Plenum Box Dimensions	Please indicate if special dimensions are requested K_E x K_B x H x ØD (mm) x s (no. of inlet spigots)

C

CDA

Louvered Face
Ceiling Diffuser

KES KLiMA
INDUSTRIAL AND TRADE CO.
Uzay Çağı Caddesi No:10
06370 Ostim/ANKARA
Phone:+90.312.385 76 57
Fax :+90.312.354 12 31
www.kesklima.com

FANS

Introduction

The fan is a rotor-dynamic machine which propels air or any other gas continuously.

The main types of fans

The fans can be divided into two main types according to their air –flow pattern.

1.Axial Fan:-

It gives high flow rate and low static pressure. Axial flow fan cause air to flow parallel to the axis of the fan. It can be subdivided to propeller type and disc type (tube axial) . It is commonly used in condensers cooling.

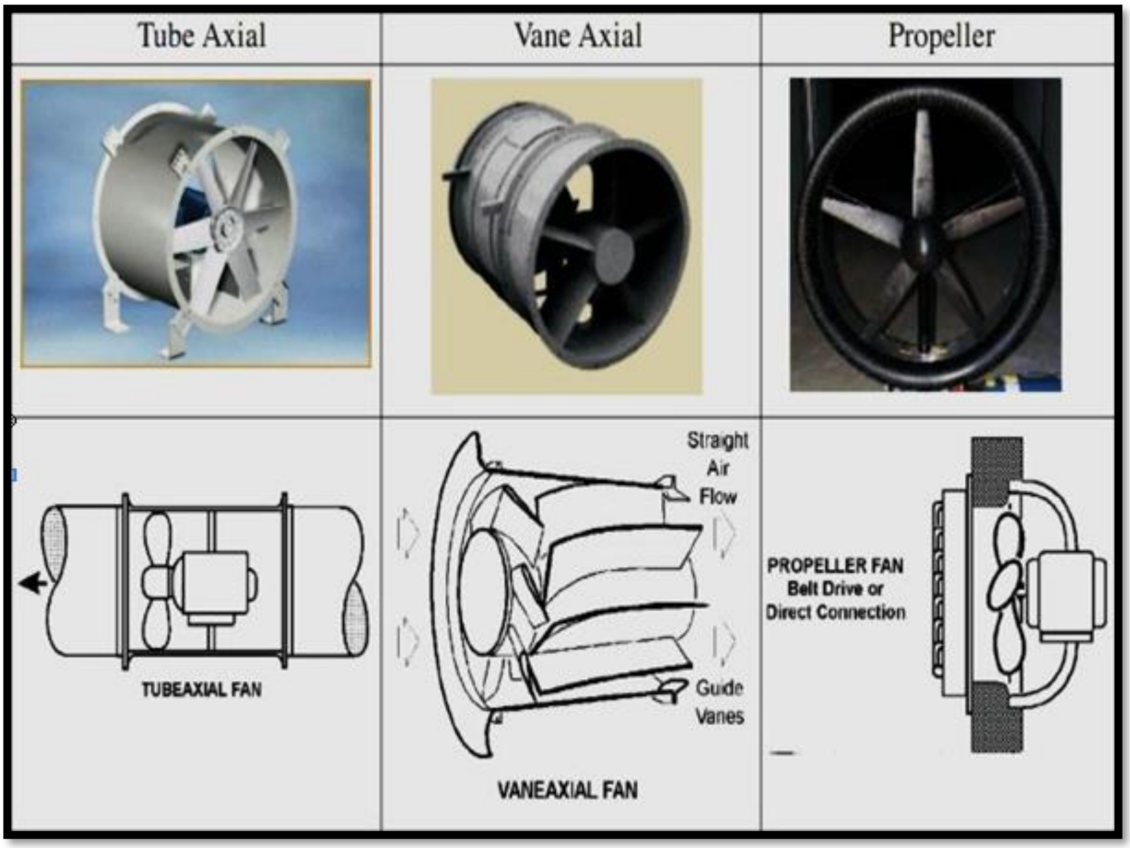


Fig. (2): Table explain characteristics of different axial fans

Type of fan	Advantages	Disadvantages
Propeller fan (Figure 11)	▪ Generate high airflow rates at low pressures ▪ Not combined with extensive ductwork (because the generate little pressure) ▪ Inexpensive because of their simple construction ▪ Achieve maximum efficiency, near-free delivery, and are often used in rooftop ventilation applications ▪ Can generate flow in reverse direction, which is helpful in ventilation applications	▪ Relative low energy efficiency ▪ Comparatively noisy
Tube-axial fan, essentially a propeller fan	▪ Higher pressures and better operating efficiencies than propeller fans ▪ Suited for medium-pressure, high airflow rate	▪ Relatively expensive ▪ Moderate

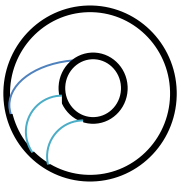
2-Centrifugal Fan:-

It is also known as radial flow fans .The air enters the fan axially and discharged radially from the fan. They were used for the duct system as it can develop considerable high pressure compared with axial flow.

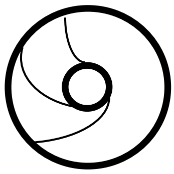
It divided into three types according to the curve of their blades relative to the impeller radii.

The two types of the blades are :

1-Forward



2-Backward



The centrifugal fans are used in most A/C systems because of its range of high efficiency.

Its characteristics is high flow rate, high efficiency and low noise and high static pressure.

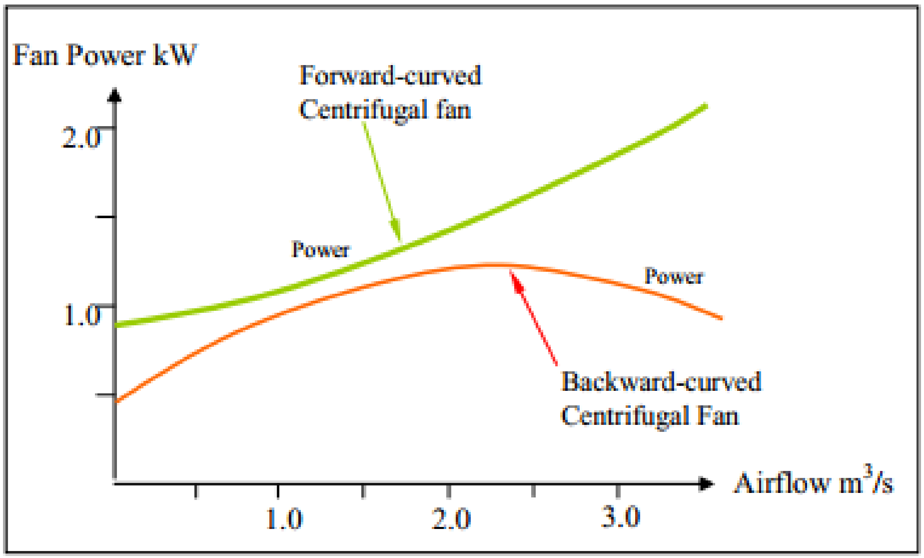
Comparison between two types:

1-The absolute velocity of Back – ware blades are less than Forward blades for some speed and impeller diameter.

2-The advantages of Forward are that provides max. air volume for a given wheel size and speed therefore, it is lighter in weight, small in size compared with other types when Q is the same.

3-F.W. blades provide greater static pressure for a given speed than the other types and it is commonly used in A/C system.

4-To obtain the same capacity under similar pressure conditions, the B.W. fans must be operated at higher speed than F.W. fan



Centrifugal Fan Characteristics

Selection of fan: To select a fan, the following data are needed:

1-Volume of air required.
2-Total static pressure, including pressure drop in filter, coils, elbows, etc.
3- Cost and efficiency.

Fans laws:

For a given fan size, ducting system and air density:

$$\frac{Q1}{Q2} = \frac{N1}{N2} \quad \ldots\ldots\ldots\ldots \quad (1)$$

$$\frac{ps1}{ps2} = \left(\frac{N1}{N2}\right)^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

$$\frac{Power1}{Power2} = \left(\frac{N1}{N2}\right)^3 \quad \ldots\ldots\ldots\ldots\ldots \quad (3)$$

Example: A fan running at (256r.p.m). Handles 8.2 m^3/s at a fan static pressure (Ps) of 250 Pascal. The fan power was (3.40 KW. If the fan speed is increased to (300r.p.m) , Calculate :

1. New air flow rate.
2. New static pressure.
3. New fan Power

Solution:

$N1=256$ rpm $N2=300$rpm
$Q1=8.2\ m^3/s$ $Q2=?$
$Ps1=250$Pa. $Ps2=?$
$Pf1=3.4$ Kw $Pf2=?$

1.$Q1/Q2=N1/N2 \rightarrow Q2=Q1*N2/N1= 8.2*300/256=9.61\ m^3/s$
2.$Ps1/Ps2=(N1/N2)^2 \rightarrow 50/Ps2=(256/300)^2 = Ps2=343.3$ Pa
3.$Pf\ 1/Pf\ 2 = (N1/N2)^3 \rightarrow 3.4/Pf\ 2= (256/300)^3 \rightarrow Ps2=5.47$ KW

H.W.
A fan running at 382 r.p.m. handles $33m^3/s$ at Ps ($3N/m^2$) and a fan power of (3KW). If the fan sped increased to (440 r.p.m.), what will be the value of:

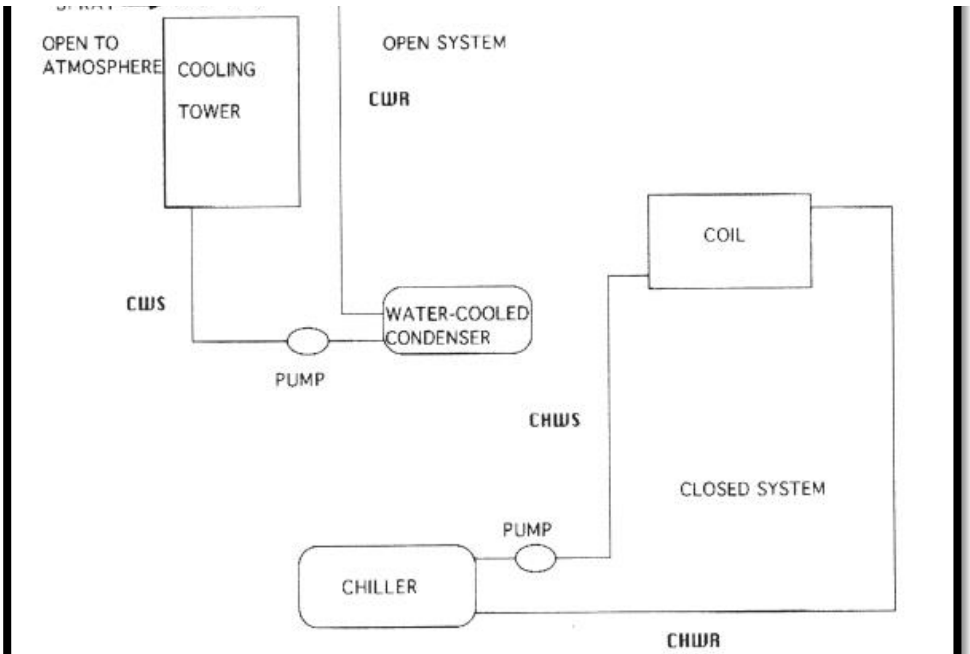
1.Fan power (new)
2.Fan static pressure (new)
3.Fan new air flow rate .

Introduction

A piping system is the means by which the thermal energy of a fluid is transported from one place to another. The type of fluid and its temperature and pressure influence and limit the choice of piping materials. Most systems are closed; i.e., the fluid is continually reticulated and no makeup water is required except to replace that lost due to leaks. Steam systems are partly to completely open as when the steam is used for a process or humidification and require continuous makeup water. Cooling-tower water systems are open and need makeup water to replace the water evaporated in the tower. Closed systems require some means of compensating for the changes in volume of the fluid due to temperature changes. Expansion (compression) tanks are used. Piping must be properly supported, with compensation for expansion due to temperature changes and anchors to prevent undesired movement.

Open and Closed Piping Systems

An open system has a break in the piping and the water is "open" to the atmosphere. A closed system has no break in the piping and the water is "closed" to the atmosphere. A typical air conditioning chiller gives examples of both the open and closed piping system. The water-cooled condenser and cooling tower loop of the air conditioning system is an open piping system; the loop from the water cooler to the chilled water coil is a closed piping system.



Open and Closed Piping Systems

Types of Pipe System Supply

1. Single Pipe (One Pipe System)

Older heating systems sometimes used a 1-pipe system of distribution.

2. Double Pipe (Two Pipe System):

Two-pipe systems are used to ensure that the water temperature to each coil is the same as the water temperature leaving the boiler or chiller. This should be the case if there are no water leaks and the piping is properly insulated. Because the supply water temperature is the same at each coil, two-pipe system can be used for any size application. Two-pipe arrangements have two mains, one for supply water and one for return water. Each coil is connected by a supply and return branch to its respective main. This design allows for separate control and servicing of each coil. The return connections from the coils can be made either direct- or reverse-return.

A three-pipe system has two supply mains and one return main. One supply circulates chilled water from the chiller(s), and the other supply circulates heated water from the boiler(s). The return main carries water from each coil back to either the chiller or boiler. The return connections from the coils can be made either direct- or reverse-return. A three-way valve at the inlet of each coil delivers either cold or hot water to the coil. The supply water streams are not mixed. When both cold and hot water are available, any coil can either heat or cool without regard to the operation of any other coil. Typically during the year (e.g., spring and fall) there are times that the HVAC system is simultaneously heating and cooling, with the return pipe carrying a mixture of both hot and cold water. The result is that both the chiller and the boiler receive warm water and must use more energy in order to supply their proper water temperature. Three-pipe systems use less piping than four-pipe systems and therefore are less expensive on initial cost, but they use more energy, resulting in greater long-term costs.

4. Four Pipe System:

A four-pipe system consists of two separate two-pipe arrangements. One two-pipe arrangement is used for chilled water; 200 HVAC Fundamentals the other is used for hot water. No mixing occurs. The return connections from the coils can be made either direct- or reverse return. The air handling unit usually has two separate water coils, one for heating and one for cooling. The water flow through each coil is controlled by either a two- or three-way modulating automatic temperature control valve.

Direct- and Reverse-return Pipe Systems

A direct-return piping system is routed to bring the water back to the pump by the shortest possible path. The heating or cooling coils are piped so that the first coil supplied is the

Balancing valves are required for flow adjustments since water will follow the path of least resistance, and the coils closest to the pump will tend to receive too much water while the coils farthest from the pump will be starved. A reverse-return piping system is designed so the length of the circuit to each coil and back to the pump is essentially equal in pressure drop. The coils are piped so that the first coil supplied is the last returned, and the last coil supplied is the first returned. Reverse-return systems generally need more piping than direct-return systems. Reverse-return systems are sometimes considered self-balancing because the intent of the design is to have equal pressure drops throughout the loop. However, because of varying circumstances in design or installation, reverse-return systems are usually not self-balancing, and balancing valves are still required for proper flow adjustments (Figure 1).

S - water source (chiller or boiler)
P - pump
C - coil
BV - balancing valve (manual).

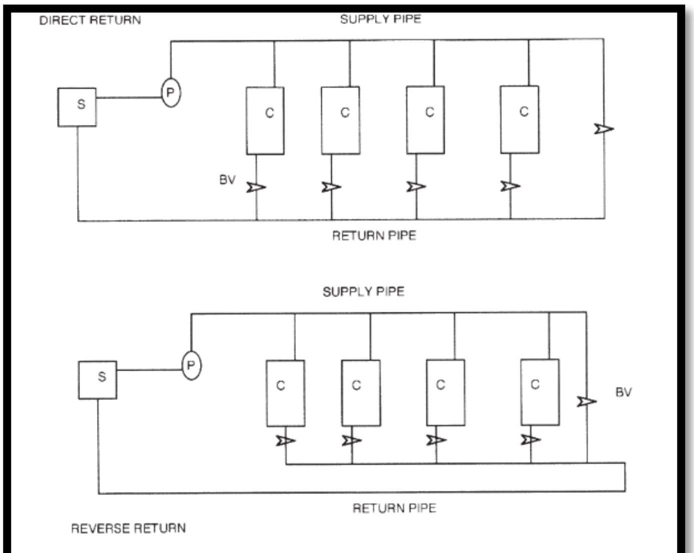


Figure 1 Direct- and reverse-return piping systems.

Types of Valve:

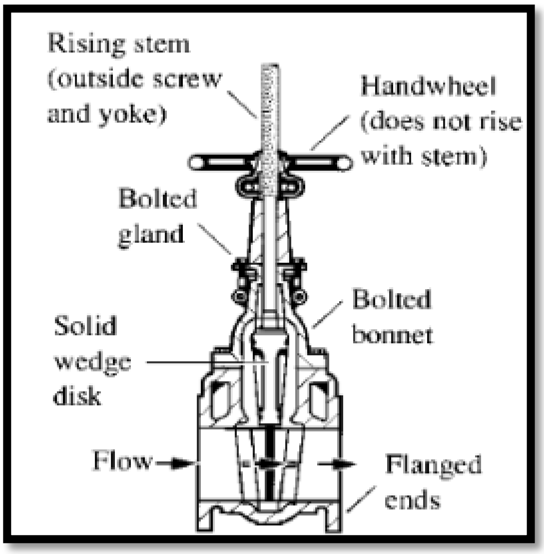
1. Gate Valves

– The disk of a gate valve is in the shape of a gate‖ or wedge.

– When the wedge is raised at the open position, a gate valve does not

– The wedge can be either a solid wedge, which is most commonly used, or a split wedge, in which two disk halves being forced outward fit tightly against the body seat.

– Gate valves are used either fully opened or closed, an on/off arrangement. They are often used as isolating valves for pieces of equipment or key components, such as control valves, for service during maintenance and repair.



2. Globe Valves:

– They are so named because of the globular shape of the valve body

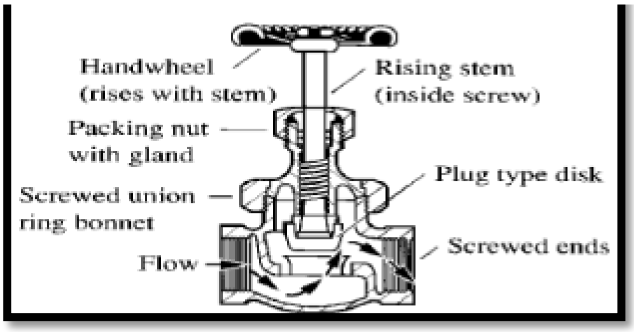
– Globe valves have a round disk or plug-type disk seated against a round port.

– Water flow enters under the disk. Globe valves have high flow resistances.

– substantially faster than gate valves. Angle valves are similar to globe valves in their seats and operation. The basic difference is that the valve body of an angle valve can also be used as a 90° elbow at that location.

– Globe valves are used to throttle and to regulate the flow.

– They are sometimes called balancing valves. They are deliberately designed to restrict fluid flow, so they should not be used in applications for which full and unobstructed flow is often required.



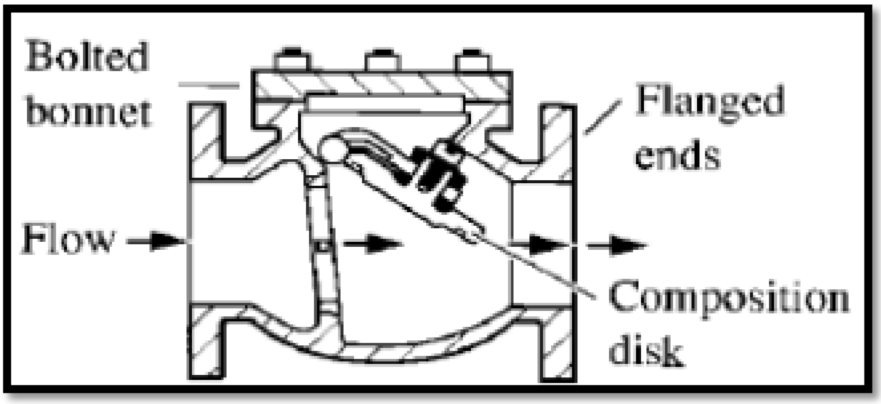
3. Check Valves:

– As their name suggests, are valves used to prevent, or check, reverse flow.

– There are basically two types of check valves: swing check and lift check. A swing check valve has a hinged disk. When the water flow reverses, water pressure pushes the disk and closes the valve.

– In a lift check valve, upward regular flow raises the disk and opens the valve, and reverse flow pushes the disk down to its seat and stops the backflow.

– A swing check valve has a lower flow resistance than a lift check valve.



A butterfly valve has a thin rotating disk. Like a ball or plug valve, it varies within a quarter-turn from fully open to fully closed. As described in Sec. 5.6, a butterfly valve exhibits low flow resistance when it is fully opened. The difference between a butterfly valve used for control purposes and a hand operated butterfly valve is that the former has an actuator and can be operated automatically. Butterfly valves are lightweight, easy to operate and install, and lower in cost than gate valves. They are primarily used as fully open or fully closed, but they may be used for throttling purposes. Butterfly valves are gaining in popularity, especially in large pipes

5. Plug Valves:

These valves use a tapered, cylindrical plug disk to fit the seat. They vary from fully open to fully closed positions within a quarter-turn. Plug valves may be used for throttling control during the balancing of a water system.

6. Ball Valves:

These valves use a ball as the valve disk to open or close the valve. As with plug valves, they vary from fully open to fully closed positions within a quarter turn. As with gate valves, ball valves are usually used for open/ close service. They are less expensive than gate valves.

1- The cooling or heating load of each room to be condition must be calculated in (Kw)

2- A sketch on the building plane should be made to show pipe runner and location at all radiators and boiler to scale in order to show the lengths.

3- The mass flow rate of water in (kg/s) must be calculated for each radiator , then summed up to obtain the total mass flow rate of the pump by using the following eq.

$Q= m.C_p. \Delta t_w$

Q: Cooling or heating load of the room (kw)

m: Mass flow rate of the water (kg/s)

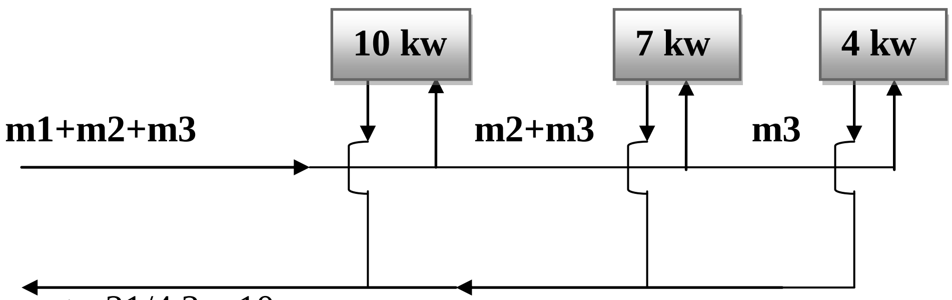
C_p: Sp. heat of the water 4.2 kJ/kg.c^o

Δt_w: Temp. difference of supply and return water it is usually (10C^o)

Example: - 3 Room having 10 kw, 7 kw, 4 kw cooling load respectively find out the cold water mass flow rate necessary to cold the rooms?

Sol.

Total cooling load = 10+7+4= 21 kw



mt= 21/4.2 x 10

mt =0.5 kg/s

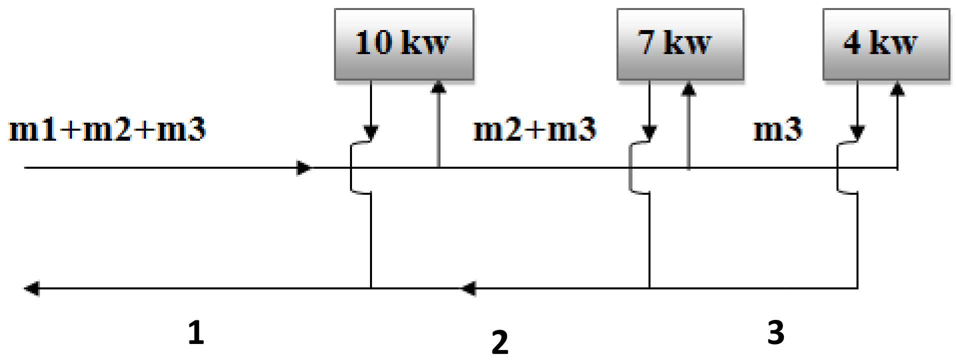
m2= 7/4.2 x 10 = 0.166 kg/s

m3= 4/4.2 x 10= 0.095 kg/s

****

4- In order to find the pressure losses in the straight run select the pipe sizes by using (piping diagram). From the diagram find out the pressure loss per meter run of the pipe by selecting the diameter (suitable)

Ex.: - select pipe size for the previous example.



Sol.

For run **number 1** both (supply &return) from piping diagram at mass rate of **0.5kg/s** and velocity **0.5 m/s** the pipe diameter is **32mm** (1.25 inch) and the pressure drop is **125pa/m** also from piping diagram

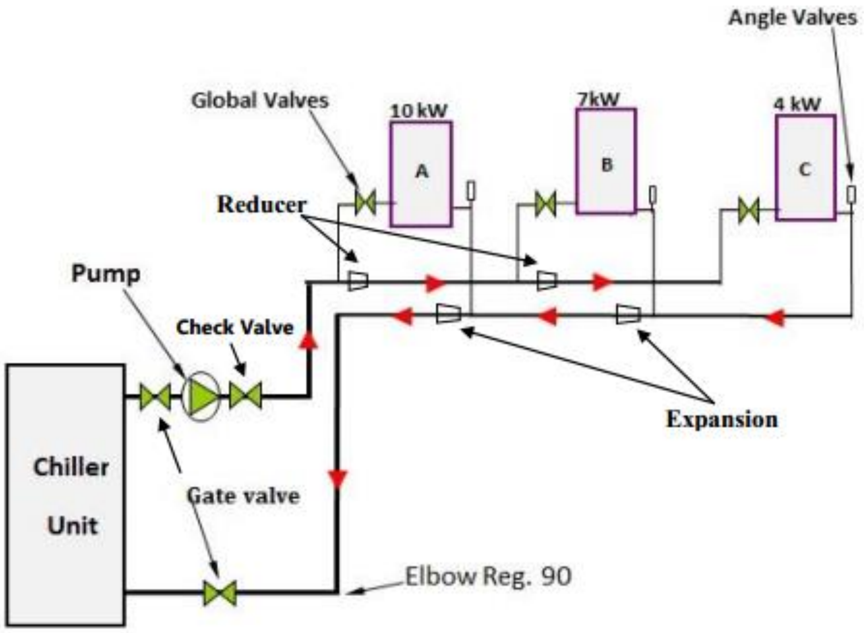
For run **number 2** both (supply &return) from piping diagram at mass rate of **0.26kg/s** and velocity **0.48 m/s** the pipe diameter is **25mm** (1 inch) and the pressure drop is **150pa/m** also from piping diagram.

For run **number 3** both (supply &return) from piping diagram at mass rate of **0.09kg/s** and velocity **0.45 m/s** the pipe diameter is **15mm** (0.5 inch) and the pressure drop is **250pa/m** also from piping diagram.

****

(elbow, tees, valves (any type), boiler, chiller and radiator) are also standard in size, available diameter 0.5 inch, 1inch, 1.25 inch, 1.5 inch… Good plan design must suggest those fitting in the diagram. as in duct an equivalent length for each fitting and for each standard size. The equivalent length of the fitting is given in table.

Example: - For the previous example find out the equivalent length of the longest run, as suggested in the following design.



The longest run is that from chiller to radiator (4kw) and return to the chiller. the following fitting are there

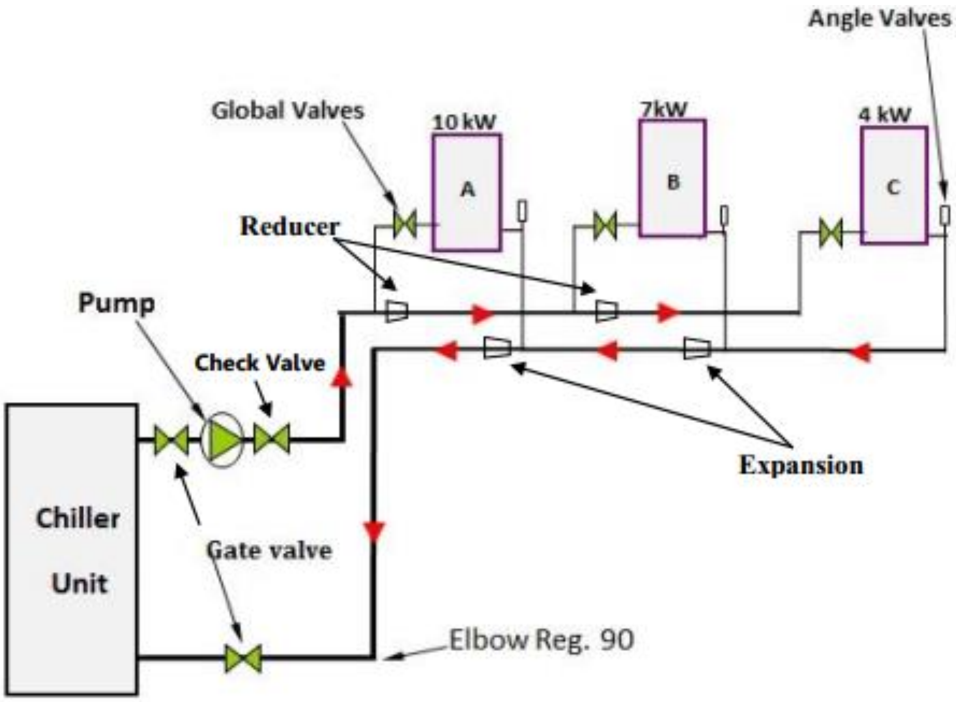
Fitting	Size inch	Equivalent length ft
Gate valve	32mm(1.25 inch)	1.3
Check valve	32mm(1.25 inch)	2
2 Elbow 90	32mm(1.25 inch)	2 x 2.6
Tee Branch	32mm(1.25 inch)	4.7
Reducer	32mm(1.25 inch)	1.0
Tee Branch	25mm(1 inch)	3.8
Reducer	25mm(1 inch)	0.8

Global valve	15mm(0.5 inch)	18.7
Radiator	15mm(0.5 inch)	4.7
Angle valve	15mm(0.5 inch)	3.6
Elbow 90	15mm(0.5 inch)	1.6
T line	15mm(0.5 inch)	2.8
Expansion	25mm(1 inch)	0.8
T line	25mm(1 inch)	3.8
Expansion	32mm(1.25 inch)	1.0
2 Elbow 90	32mm(1.25 inch)	2 x 2.6
Gate valve	32mm(1.25 inch)	1.3
Chiller	32mm(1.25 inch)	7.8

6-in order to find the overall pressure drop for each pipe size with its pipe fittings, the pressure drop is

Δp= (true pipe length (L) + equivalent for fitting length (L_e)* Δp (pa/m)

Example: - For the two pipe system mentioned in previous example find out the pressure drop (overall).



	(L)	**length (L_e)**	**length**	m	
1.25 inch	20+20= 40 m	8.99m	40+8.99= **48.99m**	125(pa/m)	**6123.75**
1 inch	10+10= 20m	2.8 m	20+2.8= **22.8 m**	150(pa/m)	**3420**
0.5 inch	10+10= 20m	10.54 m	20+10.54= **30.54m**	250(pa/m)	**7635**
					Total
					17178.75

7- How to find the pumping power

Water pump here (in two pipe system) is just for circulation (water, so there is no static head (height) even if there, it will not take in to account. The pump power way be calculated by

P= (Δp * Q*100) / η %

Where

Q: Volume rate m^3/s

ρ: density of fluid kg/m^3

m= mass flow rate kg/s

P= (17178.75 Pa *(0.5 kg/s / 998.9 kg/m^3)*100)/ 50%

Power= 1719.76 Watt

Power = 1719.76/ 746= 2.3 hp

Pump Head (m) =Total Pressure *fraction factor / (g*ρ)

Pump head = 17178.75 *10 /(9.8*998.9)

Pump head= 17.5m

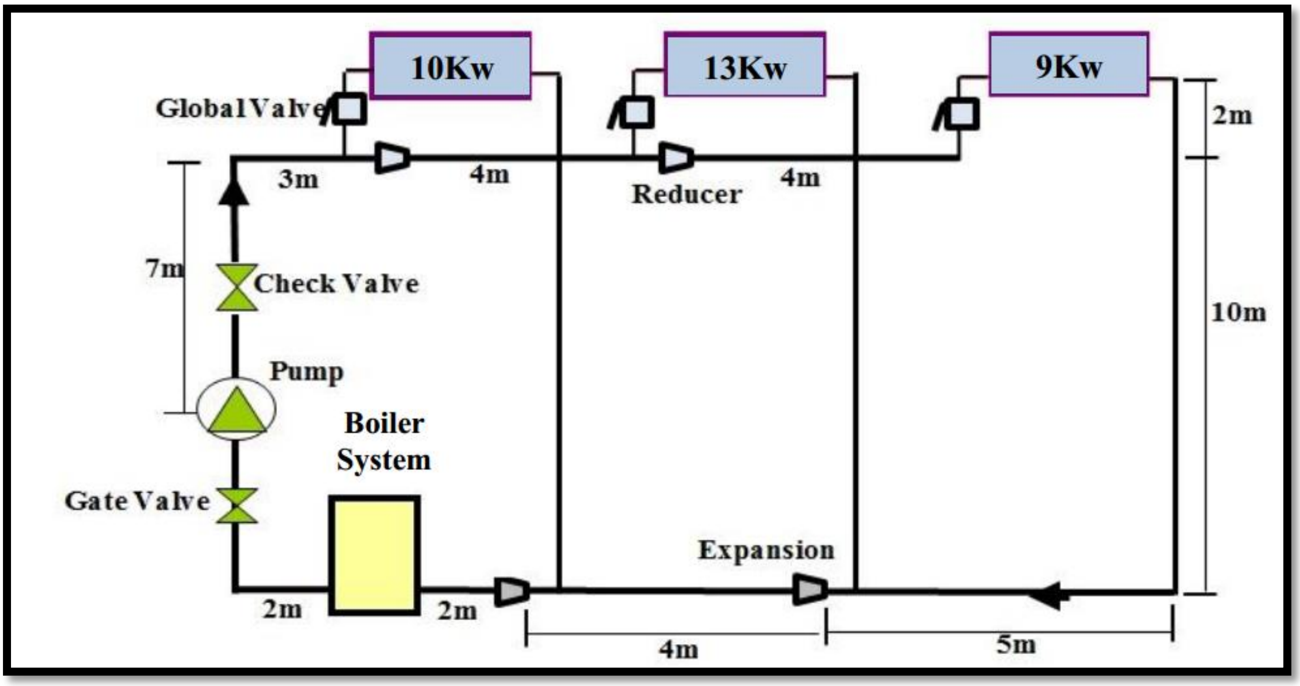
****

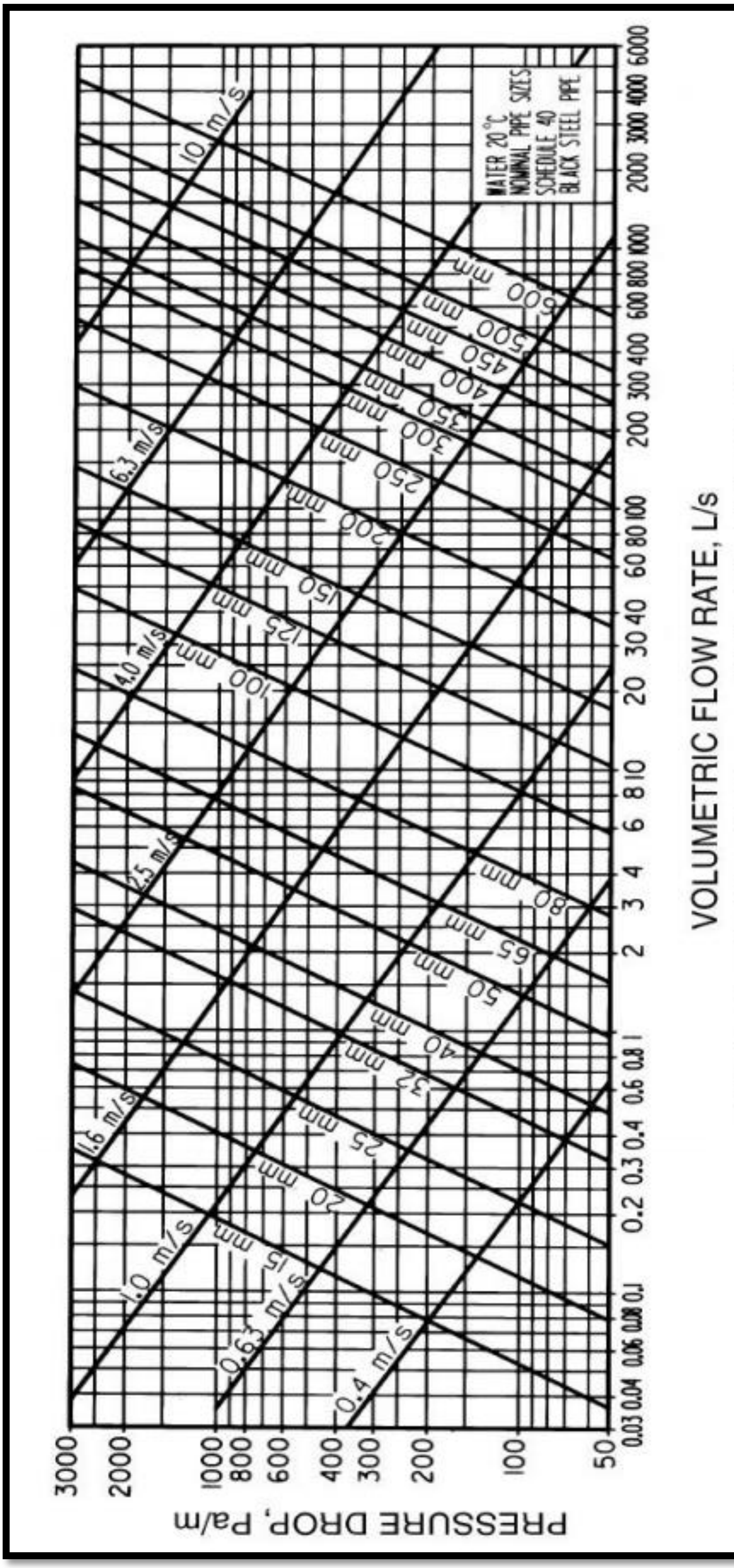
Designs the water pipe in closed system for cooling system as shown in figure below and selects the best pumps?

Note1: Assume $\Delta T=6°C$ and pump efficiency 60%.

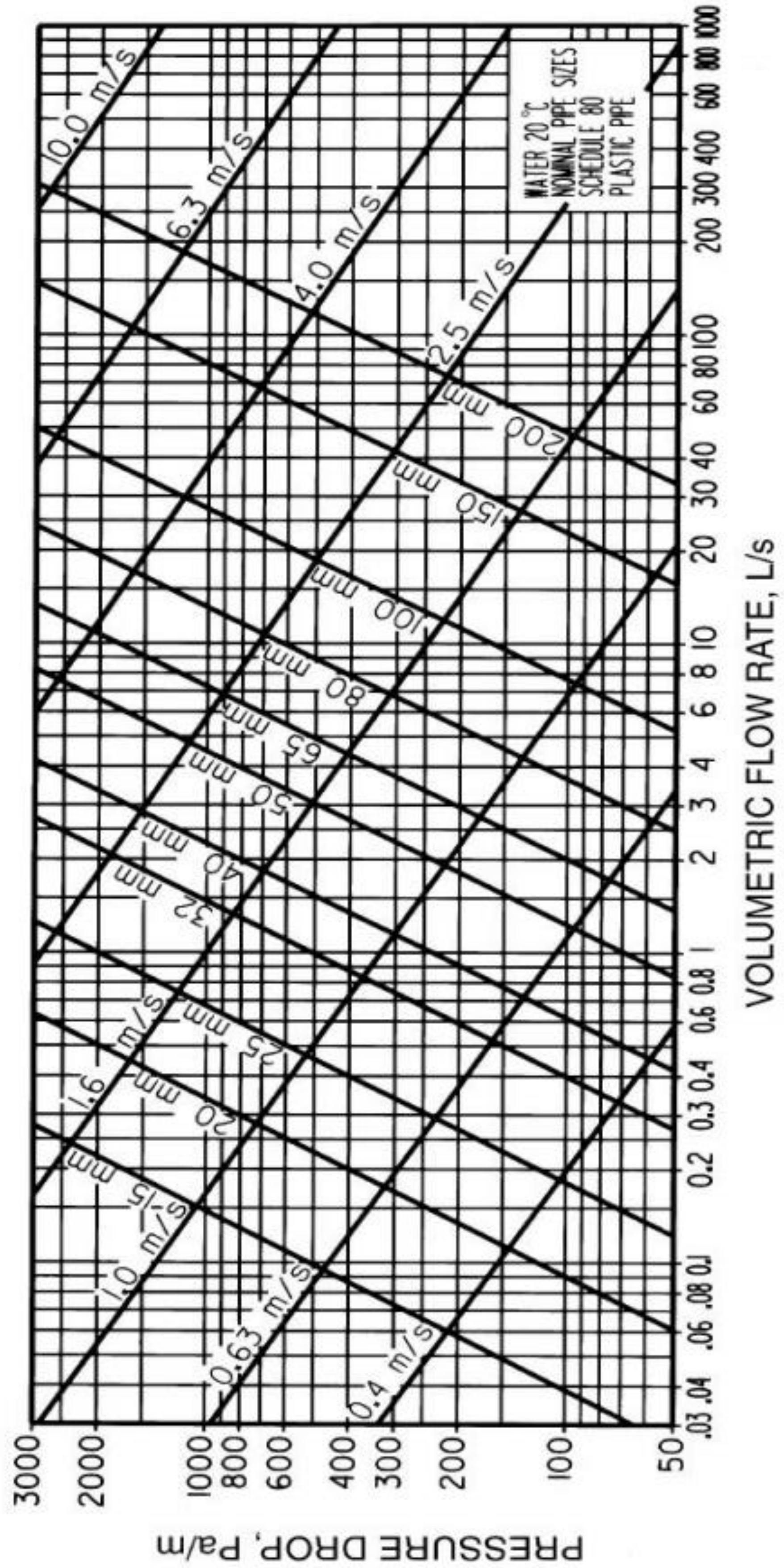
Note2: Use Heavy Grad Steel Pipe**.**

Take for water: Cp=4.2 kJ/kg. k , ρ =1000kg/ m^3





Friction Loss for Water in Commercial Steel Pipe (Schedule 40)



Friction Loss for Water in Plastic Pipe (Schedule 80)

FRICTION LOSS OF STANDARD PIPE FITTINGS

(In Equivalent Feet of Straight Pipe)

Fitting	Iron Pipe (in.)						Copper Tube (in.)				
	¾	1	1¼	1½	2	3	1	1	1¼	1½	2
Elbows											
90°	1.6	2.1	2.6	3.1	4.2	6.5	1.6	2.1	2.6	3.1	4.2
45°	1.1	1.5	1.8	2.2	2.9	4.5	1.1	1.5	1.8	2.2	2.9
90° long sweep	0.8	1.0	1.3	1.6	2.1	3.0	0.8	1.0	1.3	1.6	2.1
Tees											
100% side diversion	2.8	3.8	4.7	5.6	7.5	13	1.9	2.5	3.1	3.7	5.0
50% side diversion	6.3	8.3	10.4	12.5	16.7	25	6.3	8.3	10.4	12.5	16.7
33% side diversion	14.3	18.7	23.4	28.1	37.5	56	18.7	23.5	29.4	35.2	46.9
25% side diversion	25.0	33.3	41.6	49.8	66.7	100	31.2	41.6	52.0	62.5	83.4
Valves											
Globe (full open)	18.7	25.0	33.8	36.8	50.0	66	26.6	35.4	44.2	53.0	70.8
Gate (full open)	0.8	1.0	1.3	1.6	2.1	3.0	1.1	1.5	1.8	2.2	2.9
Stopcock (full open)	1.6	2.1	2.6	3.1	4.2	6.5	1.6	2.1	2.6	3.1	4.2
Angle (full open)	3.6	4.2	5.2	6.2	8.3	12.5	4.7	6.3	7.8	9.4	12.5
Reducer coupling	0.6	0.8	1.0	1.3	1.7	2.5	0.6	0.8	1.0	1.3	1.7
Boiler or radiator	4.7	6.3	7.8	9.4	12.5	19	6.3	8.3	10.4	12.5	16.7

What is the Pump?

Pumps are machines which supply energy to liquid in order to move it from place to another.

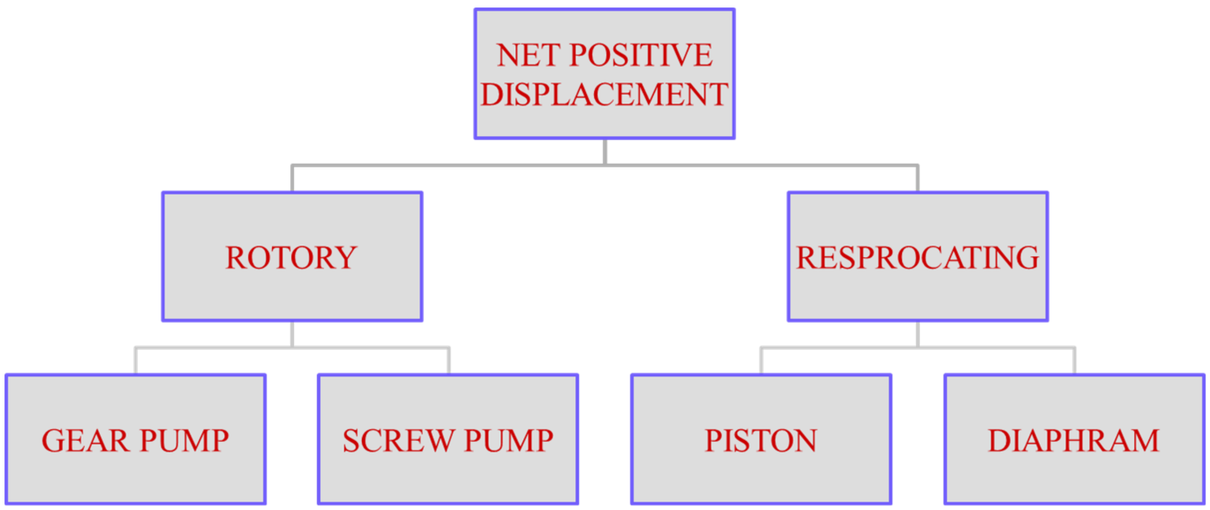
The different between the pumps and compressors is (in pump the used fluid is liquid phase but the compressors compress gases)

Pumps enable liquid to:-

1. Flow from a region or a low pressure to one of high pressure
2. Flow from a low level to a higher level

Now we can classify the pump as

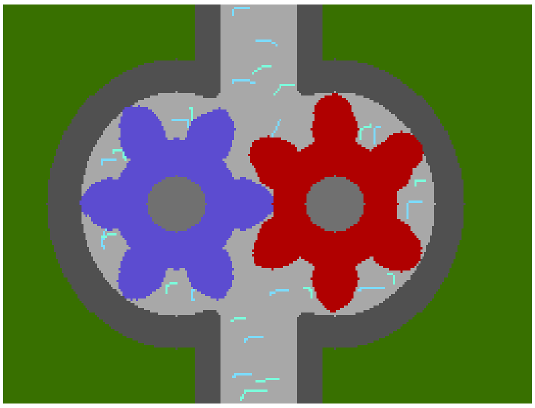
- **Net Positive Displacement**
- **Dynamic**



Net Positive Displacement

Gear Pump

Fluid is trapped between gear teeth and the housing

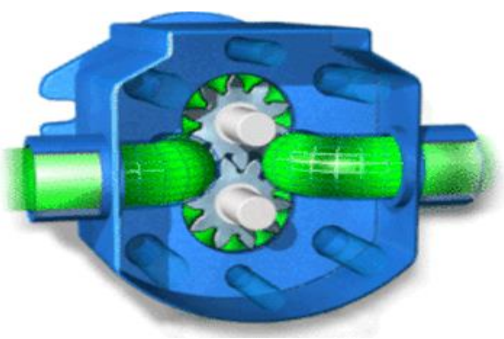


Types of Gear Pumps

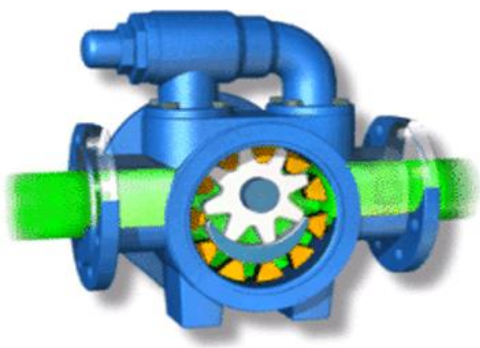
There are two main types of them.

1- one direction pumps (non reversing) these types of pumps make the fluid to flow in one way and in one direction and cant reverse the fluid direction.
2- reverse pumps these type can make the fluid flow in a certain direction and reverse its direction as we want

Reverse Type



Non Reversing Type



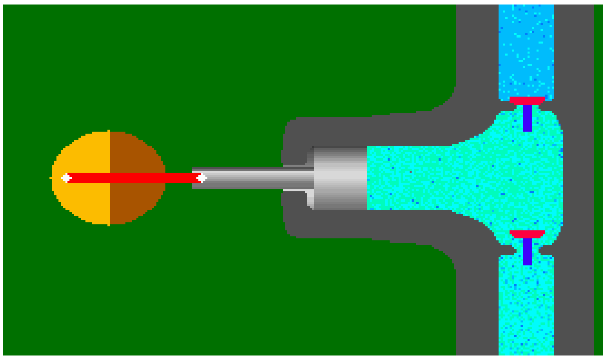
Screw Pump

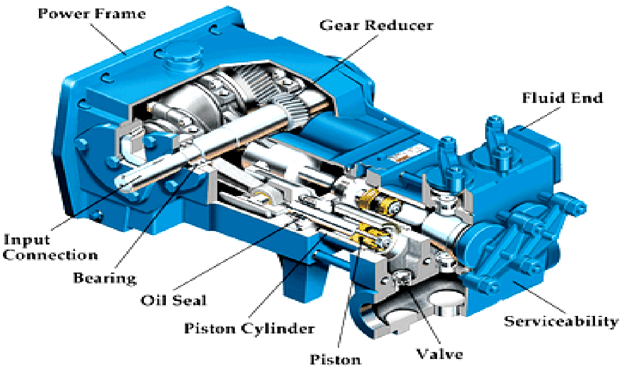
The pump forms hollow cavities which contain the fluid and move it along the screws. One screw is the drive screw and the other screw or screws is/are driven by the drive screw

Reciprocating Type

1- Piston Pump

The basic Piston Pump is very simple having just two valves and one stuffing box. In this example the reciprocating piston is driven back and forth by a rotating mechanism. This piston pump uses suction to raise water into the chamber. The lower valve can be placed below water level . The piston must be within about 25 feet of the water level, but the water can then be raised quite high.



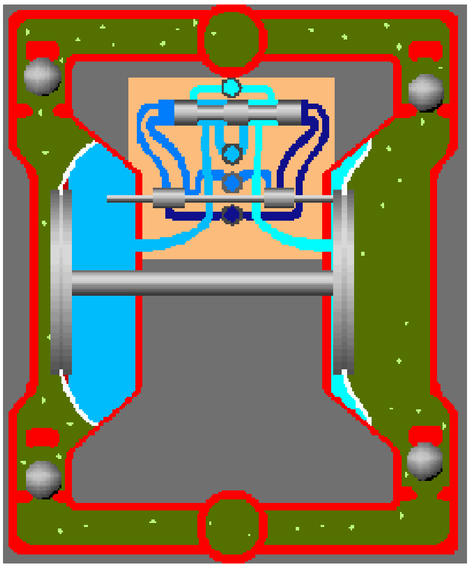


2- Diaphram Pump

- Cars often use a Diaphragm Pump to move gasoline from the gas tank to the carburetor or fuel injection plugs. The gasoline diaphragm pump in a car is operated by a cam geared directly to rotating parts of the engine. The cam pushes a pushrod
- The brown rod shown in this drawing is moved by the pushrod. It pushes the diaphragm in (a spring forces it back out)
- Fuel pumps like this one operate continuously but have a safety valve which returns fuel to the input side of the pump if pressure rises above a set level .The pump usually has a fuel filter built into it. (The fuel system will have several other filters)
- Diaphragm pumps are very common and come in many sizes. Modern plastics are flexible and long lasting making this an ideal low-maintenance pump for many applications.

Single Diaphram Pump

Double Diaphram Pump



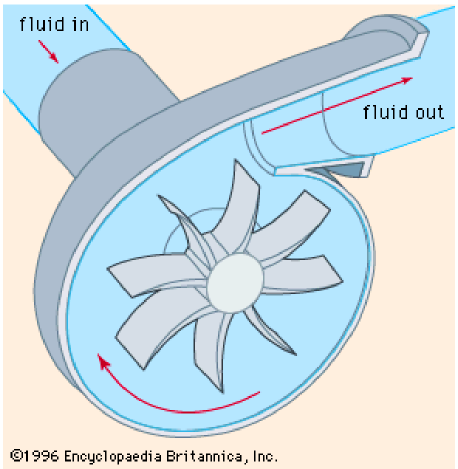
Dynamic Type

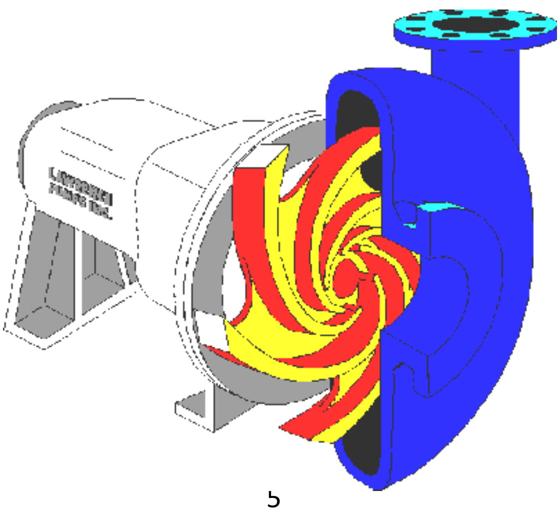
1- Centrifugal Pumps

Axial Pumps

- **RADIAL TYPE**

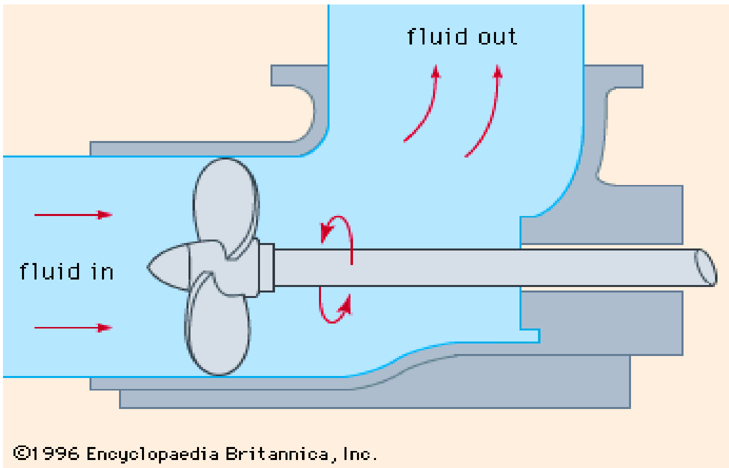
- The direction of flow is perpendicular to impeller axe
- The entry fluid must be at low point and the outlet at high point





- **Axial Type**

The direction of flow is the same direction of fan axe



Mixed Type

The flow makes an angle with the axe of impeller