भारतीय मानक

IS 18758 : 2024

Indian Standard

उत्प्रेरित प्रवात काउंटर प्रवाह कूलिंग टावरों के थर्मो-द्रवचालित डिज़ाइन — दिशानिर्देश

Thermo-Hydraulic Design of Induced Draught Counterflow Cooling Towers — Guidelines

ICS 27.060.30

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July 2024 Price Group 14

Special Structures Sectional Committee, CED 38

FOREWORD

This Indian Standard was adopted by the Bureau of Indian Standards, after the draft finalized by the Special Structures Sectional Committee had been approved by the Civil Engineering Division Council.

This standard on the [Induced draught counterflow cooling towers (IDCTs)] is published for the benefit of the entire cooling tower industry and all those who have an interest in the field.

Presently, each tower manufacturer/contractor has a different basis to arrive at a design solution to a given thermal duty using different types of fills, most of the time proprietary, and offers different cell sizes and motor capacities making it difficult for the bid evaluator to compare bids effectively. There seems to be even more confusion concerning the design aspects of cooling towers with sea water circulation. Even though there are enough guidelines in the BS codes on this subject, these are seldom referred to by the industry thus, resulting in a product incapable of meeting the desired thermal duty.

Keeping the above aspects in mind, the Committee is mandated to bring out this guideline that takes reference from various publications to compile the thermal design process in a manner that is comprehensible to anyone with an interest in the field.

It is also with a view to dispel the abstractness associated with cooling tower designs that the Committee has decided to frame certain guidelines that may be followed uniformly for the overall benefit of the industry and the environment.

This guideline applies to all such field erected custom built cooling towers that do not fall under the category of packaged or factory assembled towers and are outside the range of towers covered under CTI STD-201 and the capacity classification is not in TR.

For better understanding, solved examples for thermic design of induced draught counterflow cooling towers (IDCT) with fresh water circulation and with sea water circulation are given in Annex A and Annex B.

In the formulation of this standard due weightage has been given to international coordination among the standards and practices prevailing in different countries in addition to relating it to the practices in the field in this country. This has been met by deriving assistance from the following publications:

TP88-05 Comparative evaluation of counterflow cooling tower fills, Cooling Technology Institute, Houston

BS 4485 – 3 Water cooling towers — Part 3: Code of practice for thermal and functional design, British Standards Institution

CTI ATC 105, 2022 Acceptance test code for cooling towers, Cooling Technology Institute, Houston

This standard contributes to the UN Sustainable Development Goal 9: ‘Industry, innovation and infrastructure’, particularly its target to develop quality, reliable, sustainable and resilient infrastructure, and also promote inclusive and sustainable industrialization.

The composition of the Committee responsible for the formulation of this standard is given in Annex E.

For the purpose of deciding whether a particular requirement of this standard is complied with, the final value, observed or calculated, expressing the result of a test or analysis, shall be rounded off in accordance with IS 2 : 2022 ‘Rules for rounding off numerical values (*second revision*)’. The number of significant places retained in the rounded off value should be the same as that of the specified value in this standard.

Indian Standard

THERMO-HYDRAULIC DESIGN OF INDUCED DRAUGHT COUNTERFLOW COOLING TOWERS — GUIDELINES

1 SCOPE

1.1 This guideline covers thermo-hydraulic design aspects of induced draught counterflow cooling towers (IDCTs).

1.2 This standard excludes the provisions under Thermo-hydraulic design of natural draught counterflow cooling towers (NDCTs).

2 REFERENCES

The standards given below contain provisions, which through reference in this text constitute provisions of this standard. At the time of publication, the editions indicated were valid. All standards are subject to revision and parties to agreement based on this standard are encouraged to investigate the possibility of applying the most recent edition of these standards:

IS No.	*Title*
IS 12615 : 2018	Line operated three phase A.C motors (IE code) "efficiency classes and performance specification" (*third revision*)
IS 15999 (Part 1) : 2021/IEC 60034-1 : 2017	Rotating electrical machines: Part 1 Rating and performance (*second revision*)
IS 18705 : 2024	Thermo-hydraulic design of natural draught counterflow cooling towers — Guidelines

3 TERMINOLOGY

3.1 Air Inlet — Air inlet is the open area in the IDCT above basin kerb level through which air is sucked into the tower because of the mechanical draught.

3.2 Axial Fan — An axial fan is a propeller mounted in the middle of the fan stack to induce air flow through the cooling tower.

3.3 Cold Water Channel — Cold water channel (CWC) is an outlet from where re-cooled water flows out of the basin toward the fore-bay of a pump house.

3.4 Distribution System — Distribution system in a IDCT comprises of hot water channels/ducts, distribution pipes and spray nozzles.

3.5 Drift Eliminator —Drift Eliminators are used in cooling towers to eliminate drift, that is, water particles/droplets being carried away by the up draught of air.

3.6 Drive Shaft — A drive shaft is a tubular connector between the electric motor and the gear box, with flexible couplings.

3.7 Electric Motor — Electric motor is the driver for the fan-gear box assembly.

3.8 Fan Stack — Fan stack is an aerodynamic enclosure that is necessary to achieve fan performance in all types of induced/mechanical draught cooling towers.

3.9 Fill or Heat Transfer Media — The 'fill' or the heat transfer media is the packing inside the IDCT over which hot water is sprayed by nozzles to exchange heat with the upward draught of air.

3.10 Gear Reducer — Gear box is a speed reduction device used to decrease the input motor speed to match the desired fan speed.

3.11 Plenum — Plenum is the empty space between the top of drift eliminator packing and the fan inlet.

3.12 Rain zone — Rain Zone is the space between the bottom of fill and the water surface in the basin. Droplets of water falling through the fill are called 'rain'.

4 SYMBOLS AND UNITS

For the purpose of this standard, the following letter symbols shall have the meaning indicated against each; where other symbols are used, they are explained at the appropriate places:

A_{ai} — Air inlet area

A_c — Column area per cell

A_d — Fan area

A_e — Effective airflow area in spray zone and planum

A_{ef} — Effective fan airflow area

A_{Fill} — Effective fill area

A_h — Hub area

A_{se} — Stack exit area

BKW — Break shaft power

CL — Cell length

CW — Cell width

C_n — Number of column in a cell

C_p — Specific heat of water

D — Fan diameter

D_h — Fan hub diameter

D_{se} — Stack exit diameter

E_v — Evaporation loss

EMSL — Elevation above mean sea level

FH — Fill height or fill depth

FKW — Fan power required

G — Dry air flow rate

G_a — Dry air mass flow per unit fill area

G_m — Average wet air mass flow through fill

G_{wm} — Average wet air flow

G_{w1} — Wet air mass flow through inlet

G_{w2} — Wet air mass flow through exit

HL — Head loss

H_1 — Possible air inlet height

h_a — Enthalpy distribution

h_w — Enthalpy of water temprature

h_1 — Enthalpy of inlet (ambient) air

h_2 — Enthalpy of air exiting the fill

KaV/L — Tower characteristic or number of transfer units (NTU)

$(KaV/L)_A$ — Tower characteristic or number of transfer units (NTU) assumed

$(KaV/L)_D$ — Tower characteristic or number of transfer units (NTU) at demand

$(KaV/L)_{Fill}$ — Tower characteristic or number of transfer units (NTU) for fill

L — Water flow rate

L/G — Liquid to gas or water to air ratio

MBD — Main beam depth

m_{avf} — Wet air mass flow through fill

m_{avo} — Wet air mass flow at fill exit

ND — Nozzle depth

$N_{vh\text{-}ai}$ — Number of velocity head in air inlet

$N_{vh\text{-}dpf}$ — Number of velocity heads for pressure drop through drift eliminator, plenum and fan entrance

$N_{vh\text{-}ds}$ — Number of velocity head in distribution piping system

$N_{vh\text{-}sz}$ — Number of velocity head in spray zone

R — Range = $(T_1 - T_2)$

RH_1 — Related humidity at inlet

RH_2 — Related humidity at exit

SBD — Secondary beam depth

SP — Static pressure

SZ — Spray height/zone

TC — Tip clearance

TP — Total preserve

TPH — Available pumping head above ground level

t_{a1} — Inlet dry bulb temperature

t_{a2} — Exit dry bulb temperature

T_1 — Hot water temperature

T_2 — Cold water temperature

t_1 — Inlet wet bulb temperature (WBT)

t_2 — Exit wet bulb temperature (WBT)

V_{P1}	—	Velocity pressure across fan
V_{P2}	—	Stack exit
V_{pi}	—	Velocity pressure at fan inlet
V_{po}	—	Velocity pressure at fan stack outlet
V_{pz}	—	Velocity preassure at stack exit
V_r	—	Velocity pressure recovered
V_r	—	Velocity pressure recovery/ ratio at 70 percent venture efficiency
$1V_h$	—	One velocity head
WL	—	Water loading
x_m	—	Absolute air humidity in fill zone
x_1	—	Absolute air humidity at inlet
x_2	—	Absolute air humidity at exit
η_{fan}	—	Fan efficiency
v_{ai}	—	Velocity of air through inlet
v_e	—	Velocity of air through spray zone and plenum
v_{Fill}	—	Velocity of air through fill
v_{fan}	—	Velocity across fan
v_r	—	Velocity recovery at 70 percent venture efficiency
v_{se}	—	Stack exit velocity
ρ_{avf}	—	Density of wet air in fill
ρ_{avo}	—	Density of wet air at fill exit
ρ_{wm}	—	Average wet air density
ρ_{w1}	—	Wet air density at inlet
ρ_{w2}	—	Wet air density at exit

5 INDUCED DRAUGHT COUNTERFLOW COOLING TOWER

A counter flow cooling tower (*see* Fig. 1) is a device in which the interaction between air and water is at 180° through the fill, that is water falls downward and air moves upward through the fill because of the draught induced by a fan by mechanical means. IDCTs are also called mechanical draught cooling towers.

5.1 Configuration and Components

Induced draught counterflow cooling towers comprise of a battery of individual cells that are arranged either in a single row or back-to-back. The choice of single row or back-to-back arrangement is made based on the capacity of and space required by the cooling tower.

5.1.1 *Air Inlets*

5.1.1.1 A single row IDCT has air inlets on both sides of each cell (*see* Fig. 2). In case of a two-cell single row IDCT, the air inlets can be all around, that is, each cell can have three air inlets as the fan performance is matched. However, for single row IDCTs with more than two cells the air inlets should be two per cell (with ends closed) for optimum fan performance in all cells.

5.1.1.2 A twin row IDCT with back-to-back cell arrangement has a single air inlet for each cell (*see* Fig. 3). Air Inlet is the open face of the IDCT above the basin curb level through which air is sucked into the tower by the induced draught fan. In case of a four cell back-to-back IDCT, the air inlets can be all around, that is, each cell can have two air inlets as the fan performance is matched. However, for twin-row IDCTs with more than four cells the air inlets should be one per cell (with ends closed) for optimum fan performance in all cells.

5.1.1.3 Air inlets require sufficient open space all around the IDCT to ensure that unobstructed and uniform air flow into the tower on both sides is possible with least amount of resistance, thereby preventing the fans from undergoing an external pressure drop, (*see* Fig. 4).

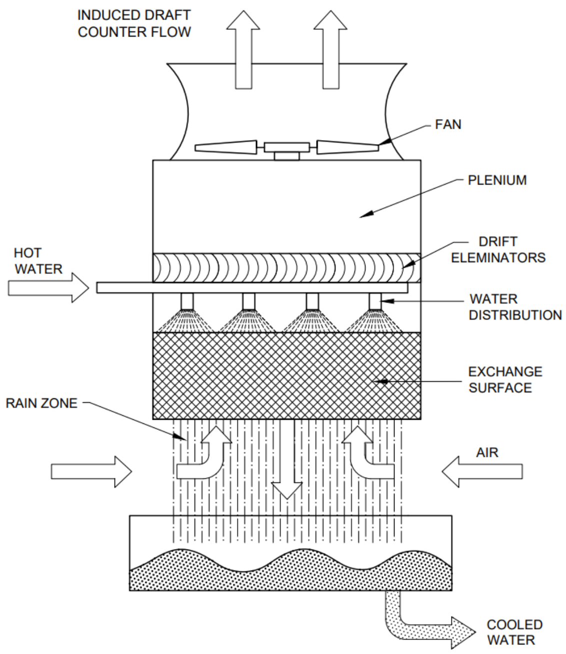


FIG. 1 INDUCED DRAUGHT COUNTERFLOW COOLING TOWER

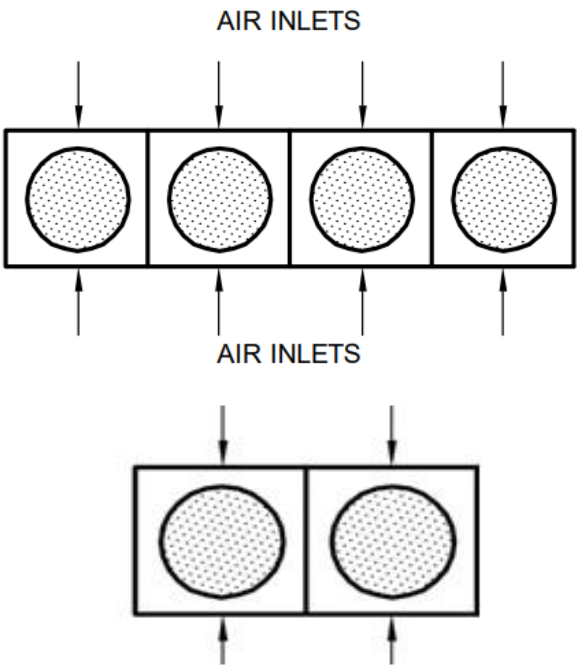
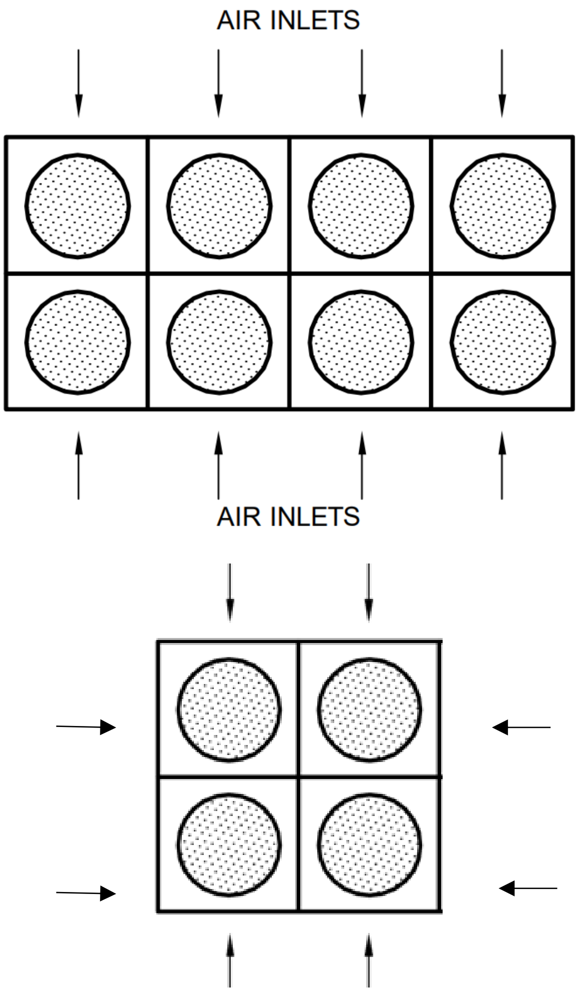


FIG. 2 SINGLE ROW IDCT



AIR INLETS (INDICATED BY ARROWS)

FIG. 3 TWIN ROW IDCT

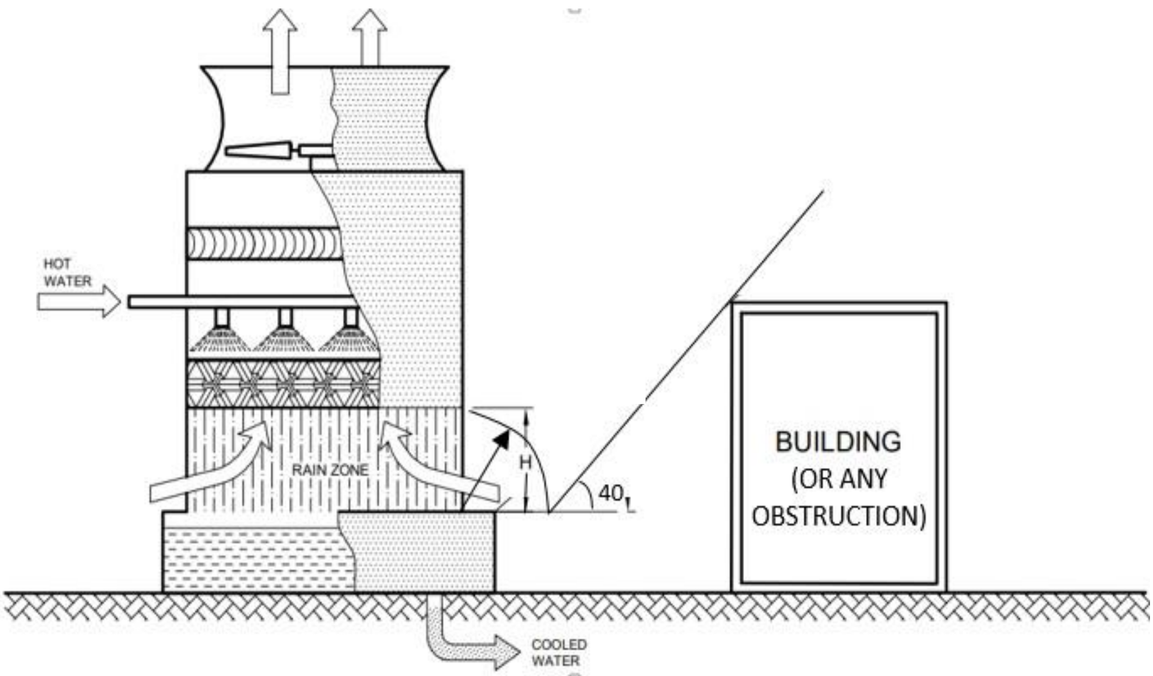


FIG. 4 SPACE REQUIREMENTS OF IDCT

5.1.4 *Rain Zone*

Rain zone is the space between the bottom of fill and the water surface in the basin. Droplets of water falling through the fill are called ‘rain’. The rain zone in IDCTs is relatively small compared to that in natural draught cooling towers.

5.1.5 *Fill or Heat Transfer Media*

The ‘fill’ or the heat transfer media is the packing inside the IDCT over which hot water is sprayed by nozzles to exchange heat with the upward draught of air induced by the fan.

5.1.5.1 Fills are mainly of three types, that is, film, splash and hybrid.

a) Film fills are a packing which form a ‘film surface’ over which water slides down either vertically or at an angle depending on the residence time and inter-mixing of water-air desired, also considering non-fouling characteristics required. Film fills usually have flutes or corrugations on individual sheets that are glued or buttoned (mechanical assembly) together to form a packing or module (*see* Fig. 5). The flute sizes usually vary between 10 mm and 33 mm and are selected based on intended applications like air-conditioning, animal farming, industrial heat transfer and power plant cold end systems.

b) Splash fills (*see* Fig. 6) come in many shapes and sizes. Some of the popular splash fills for large industrial and power plant applications in India are as under:

   1) PP splash grid;
   2) PVC Doron V bar;
   3) PVC triangular bar;
   4) RCC or PVC lath; and
   5) Wooden laths are still being used in some old process industries in medium sized IDCTs.

c) Hybrid fills are a combination of film and splash fills where the mesh structure provides surfaces for both film and droplet formation. Trickle grids (*see* Fig. 7) are an example of hybrid fills. These fills are assembled to form modules similar to film fills, thus creating a large heat transfer surface while enhancing anti-fouling characteristics due to the mesh structure.

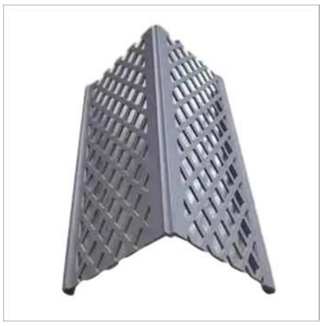
5.1.6 *Distribution System*

5.1.6.1 Distribution system in an IDCT usually comprises of hot water channels, lateral pipes and spray nozzles.

5.1.6.2 Header/riser piping delivers hot water to the hot water channel (HWC) located outside the IDCT as shown in the Fig. 8, certain variations of this distribution system are also seen in the industry. An IDCT in pultruded FRP will have the riser pipe entering inside from where the PVC laterals run across the distribution area as shown in the Fig. 9.

FIG. 5 AN ASSEMBLED CROSS-CORRUGATED FILM FILL MODULE

PP SPLASH GRID



V Bar

Triangular Bar

Wooden Laths

FIG. 6 VARIOUS SPLASH FILLS IN STAGGERED ARRANGEMENT

FIG. 7 HYBRID OR TRICKLE GRID FILL

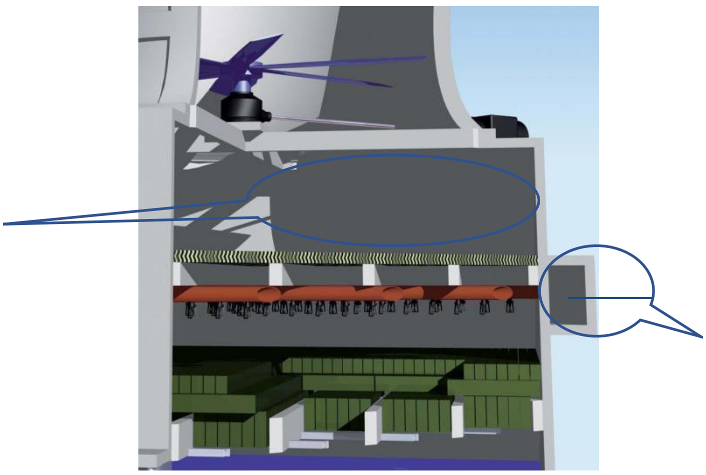


FIG. 8 IDCT CROSS-SECTION (RCC TOWER)

5.1.6.3 Distribution pipes are embedded in the HWC in a RCC tower or fixed to the riser pipe in a FRP tower to span across the width of the IDCT cell. These pipes can be fixed either above or below the supporting beams, using an appropriate fastening system, depending on the nozzle type, size and beam layout. The pressure rating and support span of the PVC pipes for any diameter shall be determined in such a manner as to limit the maximum deflection to within 0.2 percent of support span, that is, $L/500$, where L is the support span. However, for larger PVC pipe diameters (100 NB and above) where foot traffic/point loads may come into play, the bending stress may also be checked to compare with the permissible limit; in this case the live load may be considered as live UDL as such loads will be infrequent. A Solved example in this regard is presented in Annex D.

5.1.6.4 Polypropylene spray nozzles are fixed to the distribution pipes at pre-determined locations based on performance characteristics provided by the manufacturer. The typical view of the districbution system are shown in the Fig. 10 and Fig.11.

5.1.6.5 When a hot water distribution system is designed in such a way as to deliver a constant flow through each of the nozzles, it is assumed that the air and water loading is uniform over the entire fill area. However, in actual practice the air flow through the fill will vary across its width because of the resistance offered by the rain zone below fill. Because of this phenomenon the L/G varies across the cell width and as a consequence the exit air temperature too. The average L/G and exit air temperature considered in design is a result of the assumption that these variations are minimal and do not affect the results of the mathematical formulations resulting in simplified calculations.

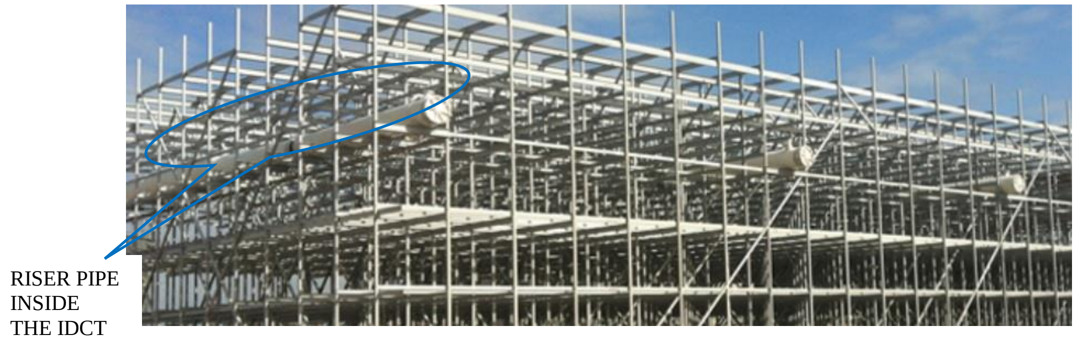


FIG. 9 INDICATIVE IDCT CROSS-SECTION (FRP TOWER)

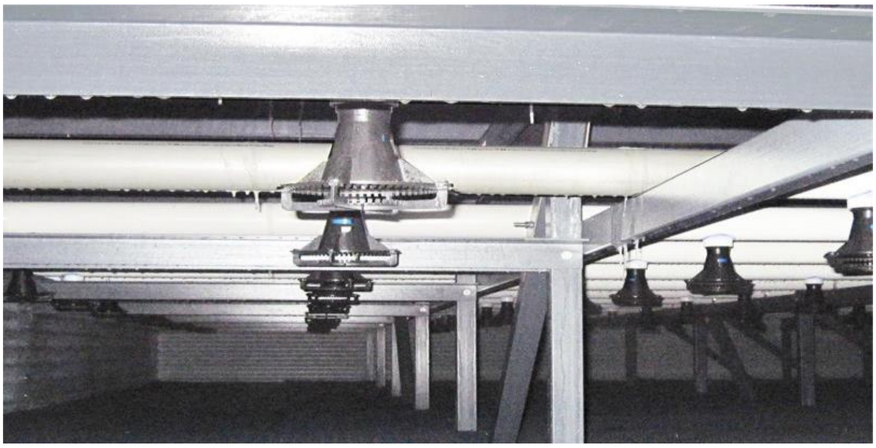


FIG. 10 INDICATIVE DISTRIBUTION PIPES PLACED ABOVE BEAMS

5.1.7 *Drift Eliminators*

Drift eliminators are used in cooling towers to eliminate drift, that is, water particles/droplets being carried away by the up-draught of air. This helps in preventing loss of water and chemicals used for water treatment and also corrosion and nuisance in the IDCT vicinity.

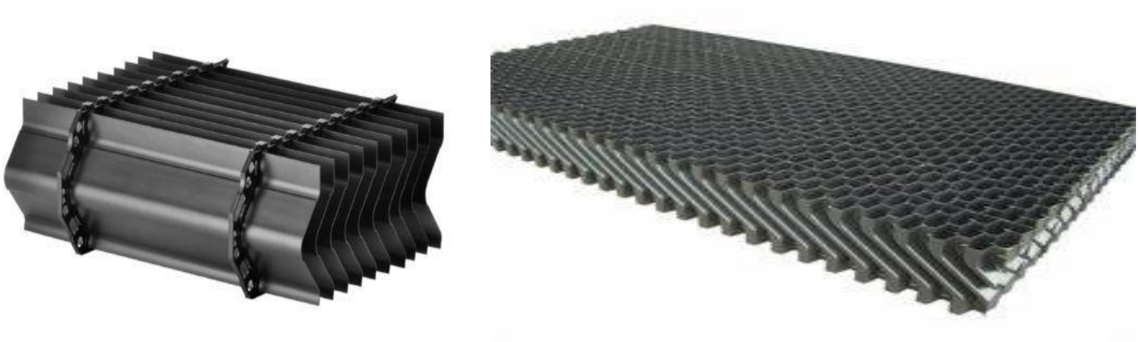
5.1.7.1 Drift eliminator packing is installed above the distribution level in case of lateral pipes fixed below the RCC beams and in between the distribution pipes in case these pipes are placed above the RCC beams.

5.1.7.2 Drift eliminators can be of blade or cellular type (*see* Fig. 12). The choice of the type of drift eliminator will depend on drift loss limitation, associated pressure drop and client specifications.

5.1.8 Plenum is the empty space spanning between the top of drift eliminator and the bottom of deck slab/roof deck. The plenum needs to be of a specified height that is adequate enough to prevent non-uniform air flow into the bell mouth of the fan stack. This is an important parameter for efficient performance of the fan.

The minimum plenum height required should be determined by projecting a circle at 45º angle from the fan cylinder opening to cover at least 80 percent of the drift eliminator area. It may be noted that a rectangular cell will require a higher plenum compared to a square cell depending on the aspect ratio used.

FIG. 11 INDICATIVE DISTRIBUTION PIPES FIXED BELOW RCC BEAMS



BLADE TYPE CELLULAR TYPE

FIG. 12 DRIFT ELIMINATOR PACKING

5.1.9 *Axial Fan*

Axial fan is a propeller mounted in the middle of the fan stack to induce air flow through the cooling tower.

5.1.9.1 Fan blades should have proper tip protection to prevent erosion and should be preferably immune to water ingress. Low design blade angles with sufficient margin (at least ± 5º) before the stall zone are preferable.

5.1.9.2 Blade tip clearance should be limited to the values given in the table below.

5.1.10 *Fan Stack*

Fan stack is an aerodynamic enclosure that is necessary to achieve fan performance in all types of induced/mechanical draught cooling towers (*see* Fig. 13).

5.1.10.1 The opening of the bell mouth of the fan stack should be at least 1.15 times the fan diameter and its height and radius (*R*) as recommended by the fan manufacturer based on fan diameter being used to eliminate high turbulence of the entering air.

However, the fan stack opening diameter may vary with the configuration chosen, that is, round, elliptical or conical inlet shapes depending on fan diameters involved; fan manufacturers may be consulted for performance measures that may be required.

5.1.10.2 A velocity recovery cone may be added to the fan stack, also considering meeting the stack exit air velocity requirement of at least 6 m/s, subject to the requirements specified at 5.5.5.

5.1.11 *Gear box*

Gear box is a speed reduction device used to decrease the input motor speed to match the desired fan speed. 1 or 2 stage reduction bevel-helical gear boxes are the preferred type for cooling tower application (*see* Fig. 14).

The preferred lubrication system for the gear box is of splash type. Forced lubrication systems shall be avoided. And the service factor to be used while selecting the mechanical rating of the gear box shall be at least 2 on the name plate motor rating of the electric motor as per American Gear Manufacturing Association (AGMA) recommendations.

It is preferred to have an anti-reverse rotation device/lock on the gear box to prevent failure of drive shaft and/or fan blades and gear box.

Sl No.	*Fan Diameter*	*Min Clearance* (mm)	*Max Clearance*	
			FRP Stack	*RCC Stack*
(1)	(2)	(3)	(4)	(5)
i)	0.915 m through 2.74 m	6.35	13 mm	0.5 percent of fan diameter
ii)	> 2.74 m through 3.35 m	6.35	16 mm	
iii)	> 3.35 m through 4.88 m	6.35	20 mm	
iv)	> 4.88 m through 12.2 m	12.7	26 mm	

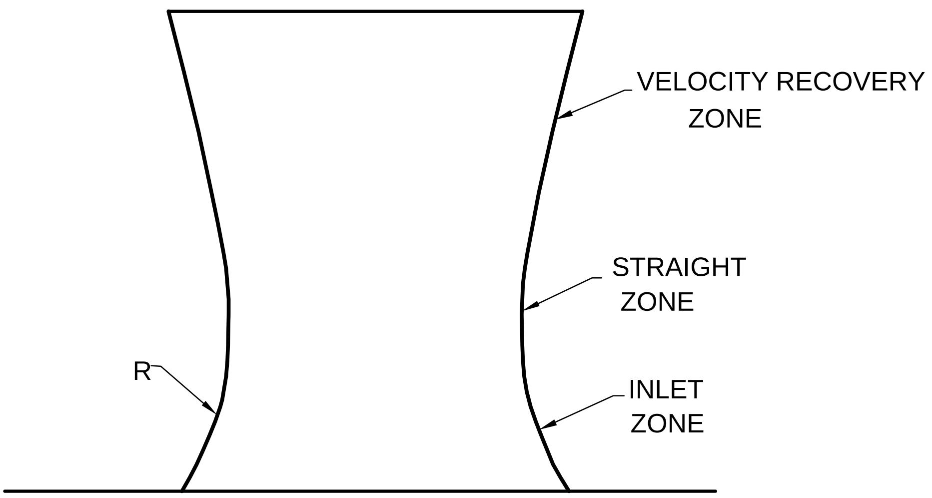


FIG. 13 FAN STACK

5.1.12 Drive shaft is a tubular connector between the electric motor and the gear box, with flexible couplings. The drive shaft can be of either stainless steel or carbon fiber material. Drive shafts in carbon fiber material will necessarily require an anti-reverse mechanism to prevent shaft failure.

5.1.13 Electric motor is the driver for the fan-gear box assembly. Usually, these motors are of single speed squirrel cage induction type with Totally Enclosed fan (TEF) cooling.

5.1.14 Butterfly valves are used in the cooling tower mainly in the riser piping for open/close operation (*see* Fig. 15). Sometimes these valves are used in the header piping also for isolation purposes. In any case these valves are not meant for flow control.

The butterfly valves in riser piping should normally be located 1 200 mm or less above local ground level for ease of operation. Higher elevations may be used if operating platforms are provided.

5.1.15 *Cold Water Channel*

Cold water channel (CWC) is an outlet from where re-cooled water flows out of the basin toward the fore bay of a pump house. The CWC will have stop log gates and debris collecting screens.

FIG. 14 GEAR BOX

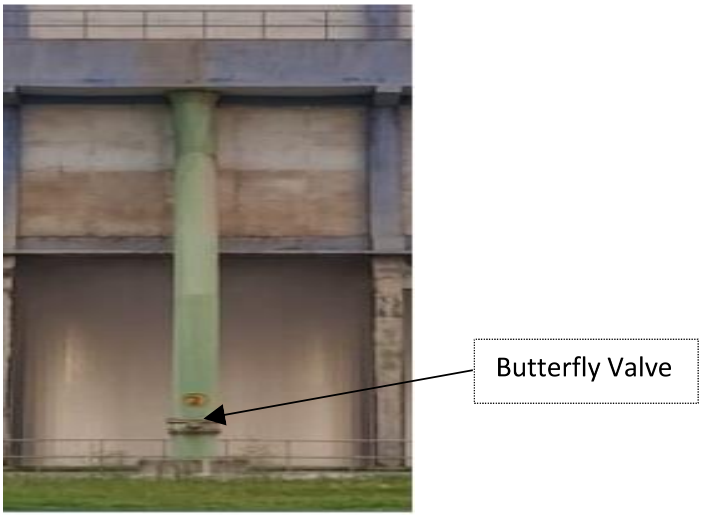


FIG. 15 BUTTERFLY VALVE

5.2 Heat Transfer Process

5.2.1 Heat transfer process in a cooling tower involves mainly the evaporation of a small quantity of water from a large circulating flow to cool it from a given hot water temperature to a specific cold water temperature. The difference between the hot and cold water temperatures is called the 'range'.

5.2.2 Since the evaporation of water occurs due to its contact with a draught of air, the cooling process is limited by the WBT of ambient air. And the driving force for this cooling process is the enthalpy difference between the air film in contact with water surface and the bulk air moving through the tower (*see* Fig. 16).

5.2.3 Many texts and references are available to study the derivation of heat and mass transfer equation of the cooling tower, which is as under:

$$KaV/L = \int_{t2}^{t1} dT \,/\, (h_{\mathrm{w}} - h_{\mathrm{a}})$$

5.2.4 Tchebycheff method given in CTI blue book and BS 4485 shall be used to solve for demand *KaV/L* using the duty parameters specified.

5.2.5 The value of *KaV/L* or the NTU is an indication of the difficulty of the cooling duty drawn in the form of 'demand curves' as shown in Fig. 17. Whereas the *L/G* ratio is taken from the intersection of the specific demand curve with the fill characteristic curve generated for a specific fill type, height and spacing or flute size.

5.2.6 The intersection of the fill characteristic curve with the demand curve gives the design *L/G*. Once the design *L/G* is established for the design of the IDCT, the heat balance is achieved using the equation given below:

$$\left(L \times C_{\mathrm{p}} \times Range\right) + (E_{\mathrm{v}} \times C_{\mathrm{p}} \times T_2) = G \times \Delta h$$

For a given tower design the tower characteristic in terms of *KaV/L* demand and design *L/G* is given as under:

$$KaV/L = C(L/G)^{-n}$$
$$or$$
$$KaV/L = CL^{-n}G^{m}$$

depending on the accuracy of results required.

5.3 Specifying Thermal Design Duty

Specifying the thermal design duty involves providing the parameters required for the design of the cooling tower. The duty parameters to be specified are as under:

a) Circulating water flow rate (m^3/h) hot water temperature (°C);

b) Cold water temperature (°C);

c) Inlet WBT (°C) (this is the sum of ambient WBT and recirculation allowance as explained in **5.3.1** and **5.3.2** below);

d) Relative humidity (in percent), if available or else 60 percent may be considered;

e) Site altitude (above MSL); and

f) Salinity of circulating water (in case of sea water) elevation of basin curb above ground level available pumping head above ground level.

Care shall be taken in specifying the pumping head as splash fills require a higher pump head than film fills because of the higher fill heights involved. Adequate pumping head must be made available for splash fill towers depending on fill heights envisaged to ensure that velocity pressure and other ratios detailed in subsequent sections are satisfied.

5.3.1 *Specifying Wet-Bulb Temperature*

Specifying the design ambient WBT requires a statistical analysis of the hourly meteorological data for at least a three consecutive year period for the project site (taken from the nearest India Meteorological Department). The design ambient WBT shall be selected in such a way that it is not exceeded by more than 5 percent of the period under consideration in a year (usually April to September).

5.3.2 *Recirculation*

5.3.2.1 Many national and international publications concerning the recirculation of the exit plume into the air inlet may be referred.

NOTE — It has been found that the CTI Bulletin PFM-110 results in a reasonable and reliable recirculation allowance than that obtained from other technical papers and research works. Hence, it is recommended that this code be used to specify the recirculation allowance for all IDCTs.

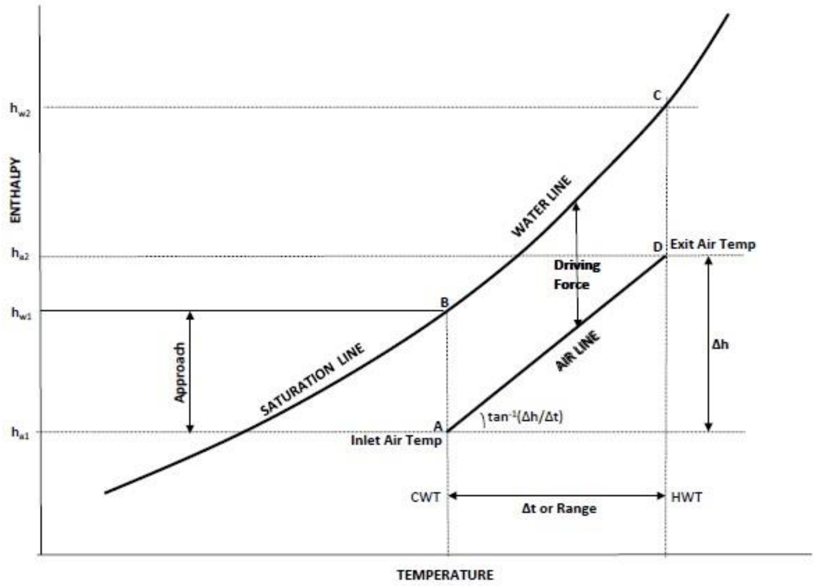


FIG. 16 HEAT TRANSFER DIAGRAM

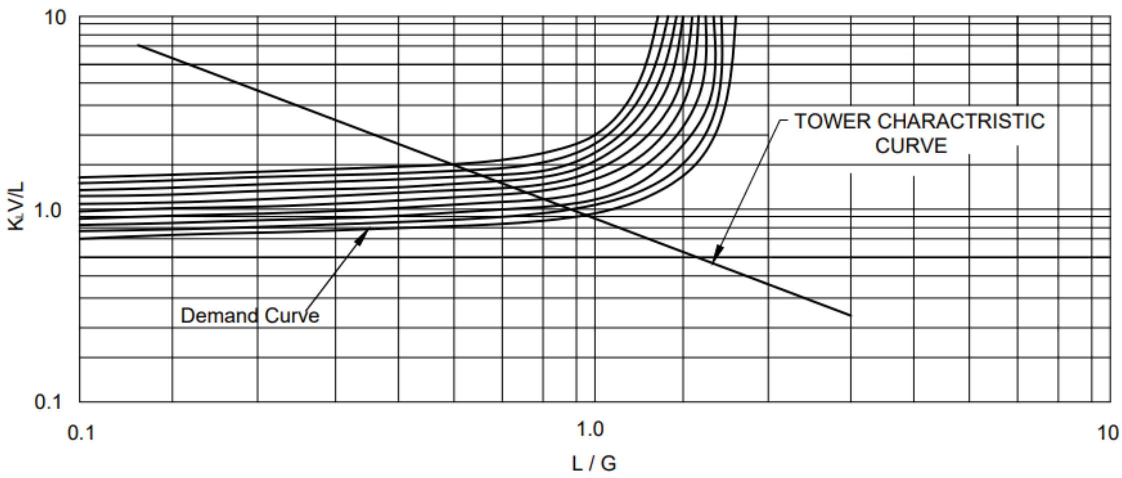


FIG. 17 TOWER DEMAND AND CHARACTERISTIC CURVE

5.3.2.2 To prevent interference in performance from adjacent IDCTs, the citing and orientation recommendations given in specified literature and relevant national and International standards may be followed. Definite recommendations cannot be made here as the requirements can vary with space, tower lengths, number of cells, stack heights, etc. Cooling tower designers shall be consulted for site/project specific recommendations.

NOTE — For specialist literatures and international standards on the subject of recirculation and interference, PFM-110, BS 4485, technical paper(s) from Marley Cooling Tower Company, Dr Detlev Kroger's book on air cooled condensers and cooling towers, etc may be referred to.

5.3.3 *Effect of Altitude on Thermal Design*

The effect of altitude on thermal design shall be considered only when the site elevation above MSL exceeds 300 m as explained in BS 4485 (*see* Fig. 18).

The available driving force increases slightly with increasing altitude (significant only beyond 300 m) resulting in either a reduced cooling tower size or fan power consumption or both.

5.3.4 *Effect of Salinity on Thermal Design*

Salinity reduces the vapor pressure and specific heat capacity and increases the density of sea water compared to pure or fresh water (*see* Fig. 19).

On the water side, in effect the increase in density of sea water is offset by the reduction in its specific heat capacity. Hence, these effects shall be ignored in thermal design. However, since the heat transfer is effected through the saturated vapor, which has a reduced pressure due to salinity, the resulting enthalpy is also reduced thereby, reducing the available driving force. Hence, corrected enthalpies shall be considered in thermal design of IDCTs for sea water application.

NOTE — See code such as BS 4485 for further reference.

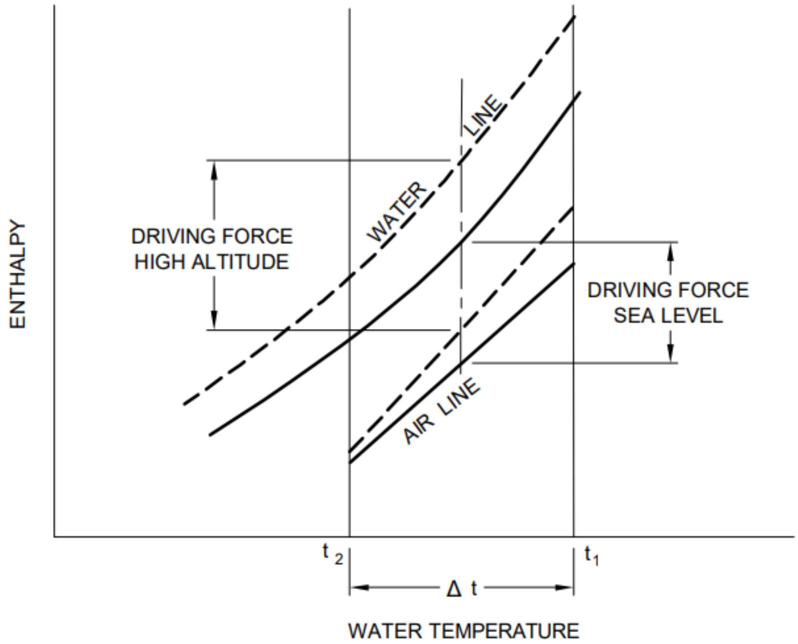


FIG.18 EFFECT OF ALTITUDE ON DRIVING FORCE

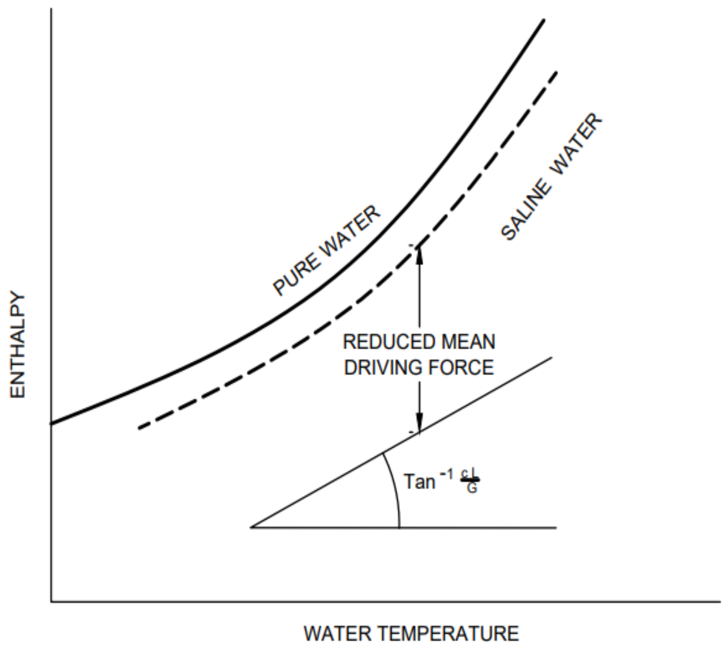


FIG. 19 EFFECT OF SALINITY ON DRIVING FORCE

5.3.5 *Specifying a Fill for Thermal Design*

5.3.5.1 Specifying a fill for thermal design involves consideration of the circulating water quality. Film fills are sensitive to water quality whereas splash fills work well with slightly poor water quality provided, there is no biological growth and excessive suspended matter.

5.3.5.2 Specific water quality requirements may be obtained from fill manufacturers based on their field performance data. In most of the cases the major source of fouling is biological growth and suspended matter. Specific slime forming bacteria that tends to stick to PVC material has been found in deposits in many IDCTs. These bacteria are of a class known as '*Pseudomonas*'. Oxidizing biocides are the most effective bio-control agents and hence, shall be used depending on water quality analysis and recommendations from the fill manufacturer and/or a specialist water treatment agency.

5.3.5.3 It shall be noted that fill selection is a compromise between thermal efficiency and fouling characteristics and hence, shall be done carefully in consultation with fill manufacturers and cooling tower designers.

5.4 Thermal Design Method

5.4.1 Thermal design of an IDCT majorly involves two considerations – one to meet the demand NTU by using a fill configuration that is capable and the other to estimate the various pressure drops for the chosen cell and fan sizes in such a way as to optimize the resulting motor power consumption and pumping head. Most often, the loading factors for power consumption and pumping head notified by the tender authority for bid evaluation purposes usually determines the design optimization. Another factor that affects optimization is the space earmarked for the IDCTs.

5.4.2 It shall be ensured that such loading factors and space availability for the IDCTs do not compromise the resulting thermal designs where the velocity pressure and other ratios as explained in subsequent sections are not met.

5.4.3 *Aspect Ratio of a Single Cell*

Aspect ratio of a single cell of an IDCT is the ratio of its longer dimension (cell length or width) to its shorter dimensions (cell length or width). Designers shall ensure that this aspect ratio is not exceeded beyond 1.25 to achieve predicted theoretical performance of the IDCTs. However, it is always preferable to have square cells in lieu of rectangular ones due to performance considerations, unless space constraints are overwhelming. An aspect ratio of 1 to 1.15 should generally, be preferable under normal conditions.

5.4.4 *Choosing a Fan Size*

5.4.4.1 Fan sizing is an important consideration to ensure that it is adequate enough to draw air from the entire area of a cell. The minimum fan size shall be chosen based on the following:

a) The fan diameter shall be at least half the longest cell dimension or the fan area shall be at least 25 percent of the cell area. For example, if the cell size is, say 16 m × 15 m, the minimum fan diameter can be:

   1) longest cell size/2 = 16/2 = 8 m ≈ 7.93 m or 8.54 m the nearest fan available sizes; and

   2) 25 percent of cell area = 0.25 × 16 × 15 = 60 m^2 that results in a fan diameter of (8.74 m) ≈ 8.54 m the nearest available size.

b) The above are the minimum criteria and both result in more or less the same fan diameter. However, larger fan diameters can be chosen for higher area coverage considering the following:

   1) Any requirement concerning the minimum space availability between adjacent fan stacks for movement of men and material (a large fan in a small cell size will make the gap between adjacent fan stacks too narrow for any movement); and

   2) In case the air velocity through fan and stack exit is high resulting in high velocity pressures adding to the fan power consumption. However, in such cases it shall be ensured that a larger fan diameter does not reduce the stack exit air velocity below 6 m/s, which is the minimum design requirement.

5.4.5 *Pressure Drops*

Pressure drop estimation requires feedback from field erected cooling towers to assess accuracy of theoretical data. The pressure drops calculations are presented in Annex C.

5.4.6 *Fan Efficiency*

The fan efficiencies shall be obtained from the fan manufacturers as different blade profiles, numbers and dimensions will result in differing fan efficiencies. However, the following table may be referred to as a guideline in case of ambiguity in vendor data.

Sl No.	*Fan Diameter* (m)	*Fan Total Efficiency* (*percent*)
(1)	(2)	(3)
i)	8.54	75
ii)	9.15	76
iii)	9.75	78
iv)	10	80

5.4.7 The double-reduction bevel-helical gear boxes used for cooling tower application have a rated efficiency of 97.5 percent. However, since a service factor of 2 is used for its selection the operating load is always close to half the rated load, which means reduced efficiency at reduced loads. In addition, there will be no load losses because of resistance to motion from lubricating oil, oil seals, gear meshing, etc. All these losses can be significant and hence, shall be considered while calculating the gear/motor shaft power. For example, if the no load loss indicated by the gear box manufacturer is 2.4 kW for a rated load of 125 kW, the shaft power, BKW will be:

$$\text{FKW/Gear efficiency} + \text{No load loss}$$

It shall be noted that the gear efficiency is different from the gear box overall efficiency. Now that BKW is established considering the no load loss, the overall gear box efficiency, η_g will be:

$$\text{FKW/BKW} \times 100$$

The ‘no load losses’ and ‘load losses’ will vary with the gear ratio and also with the manufacturer. Both these losses are inevitable and result in a lower overall gear box efficiency that must be considered while calculating BKW. Hence, it is necessary for the designer to obtain the above information from gear box manufacturers before finalizing the thermal design. And in case there is no such data forthcoming from gear box manufacturers. It is recommended that the gear box efficiency be limited to 95 percent at full load, 94 percent at three-fourths load and 93 percent at half load for all IDCT designs.

5.4.8 The motor efficiencies can easily be accessed from manufacturer catalogues. However, these are rated efficiencies presented in the catalogues and are subject to tolerance as per IS 12615 and IS 15999 (Part 1) provisions. Hence, it is recommended that the motor efficiencies be considered after accounting for the tolerance permitted as per IS 12615 and IS 15999 (Part 1) and/or cross checking with respective motor manufacturers.

5.5 Design optimization involves several checks requiring multiple thermal design runs before arriving at the best solution to achieve guaranteed performance.

5.5.1 The pumping head required for a splash fill IDCT is between 2 and 4 mWC more than that with a film fill. This is a large increase in pumping energy resulting not only in increased operating costs but also increased carbon emissions.

5.5.2 In most of the cases involving fresh or slightly brackish water applications, it may be possible to use film fills even with a necessity to replace fills once in 7 years to 8 years because of fouling/scaling as the replacement costs will still be economical compared to the increase in energy costs with splash fill. A basic water treatment program will be required for both film and splash fill applications because the first zone that gets affected in an IDCT because of poor water quality is the distribution system.

5.5.3 Splash fills have either been completely eliminated or on the fast decline in many parts of world due to the following reasons:

a) Requires higher pump head;

b) Takes longer to install;

c) Prevents easy access to fill level for maintenance of nozzles;

d) Certain types of splash fills with slotted surfaces cannot be cleaned once choked; and

e) Splash fills can sag with the passage of time affecting thermal performance.

5.5.4 The performance of all types of fills gets affected because of a poorly designed or performing distribution system, more so for a splash fill packing. Hence, the choice of fill shall be made by the designer or purchaser weighing all the options as suggested.

5.5.5 To ensure that the IDCT performs as per design it is necessary to consider the following aspects:

a) The velocity pressure ratio of the IDCT shall be greater than 5. The ratio of sum total of static pressures from air inlet to drift eliminator to one velocity head

($\rho v^2/2g$) at the air inlet is called the velocity pressure ratio. This ratio shall be the same for all cells in the IDCT;

b) The ratio of air inlet velocity to the air velocity through fill should not exceed 2.25 wherever possible (depends on pump head availability) as it affects air distribution through the tower. The higher this ratio beyond 2.25 the greater will be the fan stack exit air velocity required (beyond 6 m/s) to prevent recirculation;

c) The fan stack exit air velocity shall be at least 6 m/s or the ratio of fan stack exit air velocity to air inlet velocity should be at least 1.25 to prevent recirculation, whichever is higher. Wherever possible, this ratio should be increased to 1.5 to ensure uniform air distribution through the IDCT and also to prevent recirculation;

d) The fan diameter shall be chosen in such a way as to satisfy the requirement outlined at 5.4.4;

e) The plenum and fan stack selection shall be done in such a way as to satisfy the requirement outlined at 5.1.8 through 5.1.10;

f) Single row IDCTs should have two air inlets per cell; opening of end walls to enhance air flow through end cells is not recommended and shall be avoided. The same applies to twin row IDCTs that have a single air inlet per cell. However, exceptions to this rule are explained at 5.1.1 above;

g) The pressure drop for the wind break wall (to prevent blow out), if provided should be considered in thermal design. Certain other methods too are available in the industry to prevent blow out; and

h) Water wall-bypass can be as high as 1 percent to 5 percent in large field erected cooling towers. A water wall-bypass of about 3 percent (for properly erected fill with good workmanship) shall be considered for all splash fill designs as this phenomenon occurs invariably at all junctions with RCC members and cell walls. This consideration can be in terms of increase in design circulating water flow by 3 percent. Some bypass is bound to occur in film fill towers too, but to a much lesser extent and can be limited to about 2 percent of circulating water flow.

5.6 Hydraulic Design

Hydraulic design involves sizing of the hot water header piping, riser piping, hot water channel and distribution piping and the associated pressure drops.

5.6.1 *Hot Water Piping*

The header and the riser piping shall be sized for a velocity of 2 m/s or less as per industry practice to optimize costs. The number of TEEs, bends and reducers shall be as few as possible to reduce pressure drops in the circuit.

5.6.2 *Hot Water Channel*

Since the distribution system is usually of gravity type in an IDCT, the hot water channel is open to atmosphere and located outside the cooling tower. This channel shall be sized for a velocity not greater than 1 m/s to 1.15 m/s. Other methods of hot water distribution are sometimes implemented by contractors in the industry.

5.6.3 *Distribution Pipe Sizing*

The distribution pipes embedded in the hot water channel shall also be sized for velocities not greater than 1 m/s to 1.15 m/s. As the length of these pipes is short (< 20 m) reducers to change to smaller pipe diameters after a certain distance shall be avoided. Instead, single row IDCTs with cell widths in excess of 14 m may be provided with distribution systems on both sides.

5.6.4 *Spray Nozzle Selection*

Down spray nozzles that work efficiently with water head requirements between 0.5 m to 0.85 m shall be opted for in case of gravity type distribution system. In case of a pressurized distribution system, the nozzle head requirement may range between 1.2 and 1.5 mWC.

Up-spray nozzles, if opted for may be selected suitably based on manufacturer’s recommendations, in which case the drift eliminator packing shall be at least 1.83 m (6 feet) away from the nozzle top edge.

All nozzle selections shall be made using manufacturer’s nozzle curves that present performance in terms of water head, spray zone height and nozzle discharge diameters.

ANNEX A

(*Foreword*)

SOLVED EXAMPLE OF AN IDCT THERMIC DESIGN FOR FRESH WATER APPLICATION

Sl No.	Solved Example of an IDCT Thermic Design or Fresh Water Application				
1	Design duty parameters:				
	Water flow rate, L	= 30 000 m^3/h			
	Hot water temperature, T_1	= 43 °C			
	Cold water temperature, T_2	= 33 °C			
	Wet bulb temperature, t_1	= 28 °C			
	Relative humidity, RH_1	= 50 %			
	Salt content (in make-up water)	= 0			
	Elevation above main sea level, *EMSL*	= 0 m			
	Available pumping head above ground level, *TPH*	= 10.7 m (say)			
NOTE — Change in air properties may be considered in design only when EMSL is above 300 m (Refrence BS 4485/BSEN14705).					
2	The water flow rate is not increased in this solved example by the wall-bypass factor of 2 percent to 3 percent and the recirculation allowance is calculated as per PFM-110 that is recommended in the guideline. Hence, an appropriate decision in this regard shall be taken based on end user specifications. The above thermal duty demands a specific *KaV/L* value that can be made available by the fill and the rain zone where heat transfer takes place. *KaV/L* gain in the spray zone is ignored in this example but a decision in this regard may be taken by the users as per recommendations in the guideline.				
	The $(KaV/L)_D$ demand can be calculated using the Chebychev/Tchebycheff method outlined in ATC-105 or BS 4485, etc.				
	The solution to the problem lies in finding out what fill height (rain and spray zones are ignored here) provides the *KaV/L* demanded by the thermal duty. However, the problem can only be solved when the exit air temperature is known so that the design liquid to gas ratio *L/G* can be determined. Obviously, this warrants iterative calculations to arrive at the correct exit air temperature and the consequent *L/G*. However, it may be noted that there are several possible methods of solving this thermal problem. One such method is explained below:				

Sl No.	Solved Example of an IDCT Thermic Design or Fresh Water Application				
3	Fill selection for *KaV/L* calculation:				
	Assume marley fill MC75 film fill suitable for fresh water application				
	Fill height, *FH*	= 1.8 m (say)			
	The performance equations of this fill presented at **3.4.3.4** of the guidelines are as under:				
	$(KaV/L)_{\text{fill}}$	$= 1.035 \times (L/G)^{-0.781} \times (3.28 \times FH)^{0.584}$			
	The air inlet height is derived from the available pumping head as under:				
	Ground level to basin sill	= 0.3 m			
	Main beam depth, *MBD*	= 0.5 m (say)			
	Secondary beam depth, *SBD*	= 0.25 m (say)			
	Fill depth, *FH*	= 1.8 m or 6 feet			
	Spray height/zone, *SZ*	= 0.6 m (say)			
	Nozzle depth, *ND*	= 0.2 m (say)			
	Head required for nozzle	= 0.85 m (say)			
	Head loss in HW piping + distribution system	= 1.2 m (say)			
	Sub-total head loss, *HL*	= 5.7 m			
	Possible air inlet height, $H_1 = TPH - HL$	= 5 m			
	As the rain and spray zones are ignored, $(KaV/L)_A$	$= (KaV/L)_{\text{fill}}$			
	This $(KaV/L)_A$ will have to be matched with $(KaV/L)_D$ to determine the exit air temperature and the design *L/G*				
4	The heat balance equation of a cooling tower is $(L \times C_p \times \text{Range}) + (E_v \times C_p \times T_2) = G \times \Delta h$				
	where E_v = evaporation loss (kg/h)	E_v = loss can be calculated once inlet and outlet air properties are established			

Sl No.	Solved Example of an IDCT Thermic Design or Fresh Water Application				
	Specific heat (C_p)	= 1 (kcal/kg/°C)			
	Δh = inlet enthalpy – outlet enthalpy	= (h_1 – h_2) kcal/kg			
	Range = (T_1 – T_2)	= 10 °C			
	G = Dry air flow rate (kg/hr)				
	Solving this equation without knowing what the *L/G* and exit air temperature would be is a little cumbersome and requires an iterative process. The CTI ATC-105 heat balance equation ignores the heat of evaporated water for the sake of simplifying the calculations. However, in actual practice where design performance is guaranteed, this heat from the evaporated water cannot be ignored and hence, iterative calculations are required to be carried out. Equations are readily available for the calculation of air properties in BS, ASHRAE, DIN, Kroger, etc publications. However, air properties using equations in Kroger's book are used here for demonstration purposes as manually readings values from tables is extremely laborious. Though the CTI tables can be used for IDCT design assuming that the inlet RH is 100 percent, air properties at the inlet relative humidity will be insisted upon by clients in data sheets. It may be noted that the $(KaV/L)_D$ value varies with the tables/equations from different publications because of the datum differences. However, a uniform method of calculation based onany of these publications in their entirety will result in more or less comparable IDCT sizing and fan power consumption.				
5	Inlet air properties (from equations):				
	Design inlet *WBT*, t_1	= 28 °C			
	Relative humidity, RH_1	= 50 percent			
	Dry bulb temperature (*DBT*), t_{a1}	= 37.14 °C			
	Wet-air density, ρ_{w1}	= 1.123 9 kg/m^3			
	Abs humidity, x_1	= 0.020 2 kg/kg			
	Enthalpy, h_1	= 21.366 kcal/kg			

Sl No.	Solved Example of an IDCT Thermic Design or Fresh Water Application				
6	For a conservative design, it shall be assumed that the exit air is fully saturated, that is, its *RH* is 100 percent.				
	Now, assume an exit air temperature of say, $[(T_1 + T_2)/2] = (43 + 33)/2 = 38$ ºC for the first iteration				
	Exit air properties (from equations):				
	WBT (t_2) @ 100 percent *RH*	= 38 °C			
	Relative humidity, RH_2	= 100 %			
	Dry bulb temperature (*DBT*), t_{a2}	= 38 °C			
	Abs humidity, x_2	= 0.043 8 kg/kg			
	Enthalpy, h_2	= 36.045 kcal/kg			
	Now rearrange the heat balance equation for *L/G* as under:				
	$L/G = (\Delta h - (x_2 - x_1) \times C_p \times T_2)/(C_p \times \text{Range})$				
	Substituting values, we get *L/G*	= 1.390			
	Now, as the *L/G* and exit air enthalpy are both known, the $(KaV/L)_D$ can be calculated using the Tchebycheff method				
	NOTE — For a detailed description of the Tchebycheff method, ATC-105 or BS 4485 codes may be referred.				
	As per the Tchebycheff method all enthalpies are considered @ 100 percent *RH*				
IterationNo. 1	**Water Temperature Distribution (between T_1 and T_2)**	**Enthalpy at Water Temperature (h_w)**	**Enthalpy Distribution (h_a)**	**($h_w - h_a$)**	**1/($h_w - h_a$)**
	33		21.366		
	34	29.418	22.757	6.661 2	0.150 12
	37	34.268	26.927	7.341 2	0.136 22
	39	37.908	29.708	8.199 8	0.121 95
	42	44.071	33.878	10.192 7	0.098 11
	43		35.269		
				$\Sigma 1/(h_w - h_a)$	0.506 40
	As per Tchebycheff method, $(KaV/L)_D = \text{Range}/4 \times \Sigma 1/(h_w - h_a) \times C_p$ Therefore, $(KaV/L)_D$			= 1.266	

Sl No.	Solved Example of an IDCT Thermic Design or Fresh Water Application				
	Now, let's check how well $(KaV/L)_A$ meets the $(KaV/L)_D$ calculated				
	We have, $(KaV/L)_A = 1.035 \times (L/G)^{-0.781} \times (3.28 \times FH)^{0.584}$				
	Substituting L/G value, we have, $(KaV/L)_A$	= 2.257 1			
Iteration No.2	As the $(KaV/L)_D$ & $(KaV/L)_A$ do not match, the Exit Air Temperature will have to be iterated until a close match is achieved				
	Exit air properties (from equations):				
	Wet bulb temperature (WBT), (t_2) @ 100 percent RH	= 40.66 ºC			
	Relative humidity, RH_2	= 100 %			
	Dry bulb temperature (DBT), t_{a2}	= 40.66 ºC			
	Wet-air density, ρ_{w2}	= 1.092 5 kg/m^3			
	Abs humidity, x_2	= 0.051 0 kg/kg			
	Enthalpy, h_2	= 41.206 kcal/kg			
	Substituting values, we get L/G	= 1.882			
	Water Temperature Distribution (Between T_1 and T_2)	**Enthalpy at Water Temperature (h_w)**	**Enthalpy Distribution (h_a)**	**($h_w - h_a$)**	**1/($h_w - h_a$)**
	33		21.366		
	34	29.418	23.249	6.169 0	0.16210
	37	34.268	28.896	5.372 4	0.186 14
	39	37.908	32.661	5.246 6	0.190 60
	42	44.071	38.308	5.762 9	0.173 52
	43		40.191		
				$\Sigma 1/(h_w - h_a)$	0.7123 6
	As per Tchebycheff method $(KaV/L)_D$ = Range/4 × $\Sigma 1/(h_w - h_a) \times C_p$ Therefore, $(KaV/L)_D$			= 1.781	

Sl No.	Solved Example of an IDCT Thermic Design or Fresh Water Application				
	Now, let's check how well $(KaV/L)_A$ meets the $(KaV/L)_D$ calculated				
	Substituting L/G value, we have, $(KaV/L)_A$	= 1.781			
	The iterations will end here as $(KaV/L)_A$ meets the $(KaV/L)_D$ calculated at the assumed exit air temperature and the air properties established in this iteration will be used for further calculations.				
7	The rest of the calculations are simple as there are no more iterations to be performed. The number of cells and individual cell dimensions depend on the available space for the IDCT and the optimization of fan power consumption and/or overall cost of the IDCT will depend on evaluation criteria specified in tenders that vary with client. For this solved example, let's assume that the maximum space available is 130 m (L) × 15 m (W) excluding space required for staircases, CW channel, etc and that the minimum number of cells to be provided is eight working plus one standby.				
	As the minimum number of operating cells required is eight, it is possible to design the IDCT with more number of cells with back-to-back orientation. Optimization between back-to-back and single row cell arrangement will require lot many design runs and application of loading factors (on power and initial cost), which users of this guideline may carryout independently. Let's assume that the single row option with eight working plus one standby cell is the most optimal for this solved example.				
	Total available tower length	= 130 m			
	Assuming 500 mm wide columns in air inlet and 150 mm thick cell partition walls, space occupied by walls and end columns	= 2 m (approx)			
	Actual length available of IDCT design	= 128 m			
	Total number of cells	= 9			
	Cell length, *CL*	= 14.2 m (say)			
	Similarly, actual available Cell Width, *CW*	= 14.7 m (say)			
8	Water flow per operating cell	= 3 750 m^3/h			
	Dry air flow per cell, G_a	= $L/(L/G)$			
	Therefore, G_a	= 1 992 120.769 kg/h			
		= 553.367 kg/sec			

Sl No.	**Solved Example of an IDCT Thermic Design or Fresh Water Application**				
	Wet airflow through inlet, G_{w1}	$= G_a \times (1 + x_1)/\rho_{w1}$			
		$= 502.304$ m^3/sec			
	Average wet air density, $\rho_{wm} = (\rho_{w1} + \rho_{w2})/2$	= 1.108 240 217			
	Absolute humidity in fill zone, $x_m = (x_1 + x_2)/2$	= 0.03561			
	Average wet air flow through Fill, $G_{wm} = G_a \times (1 + x_m)/\rho_{wm}$	= 1861 571.114 m^3/h			
		= 517.10 m^3/sec			
	Average wet air mass flow through fill, m_{avf}	= 573.07 kg/sec			
	Wet air flow above fill or through exit, G_{w2}	$= G_a \times (1 + x_2)/\rho_{w2}$			
	Therefore, G_{w2}	= 532.33 m^3/sec			
	Wet air mass flow above fill or through exit, m_{avo}	= 581.59 kg/sec			
9	We can now establish the pressure drops in various zones of the IDCT using equations given **3.4.3** in the guideline:				
a)	Air inlet area, A_{ai}	$= (CL$ – End column width) $\times H_1 \times 2$ sides			
		= 137.222 m^2			
	Air velocity through inlet, v_{ai}	$= G_{w1}/A_{ai}$			
		= 3.661 m/s			
	Number of velocity heads in air inlet as per TP-88 curve, $N_{vh\text{-}ai}$	= 3			
	Therefore, air inlet pressure drop	$= N_{vh\text{-}ai} \times \rho_{w1} \times (v_{ai})^2/2g$			
		= 2.303 mmWC			
b)	Fill pressure drop equation	$= (0.000\,000\,043\,4 \times (196.8 \times v_{fill})^{2.355\,9} + 1.472\,34 \times WL \times 0.000\,000\,811 \times (196.8 \times v_{fill})^{1.540\,3} \times (1 + 0.283 \times 3.28 \times FH) \times \rho_{wm}/1.2$			
	Gross fill area	$= CL \times CW$			
		= 209.067 m^2			

Sl No.	Solved Example of an IDCT Thermic Design or Fresh Water Application				
	There will be no fill in the cell wherever columns exist. Assume 6 percent column and beam obstructions for this solved example, meaningthe effective fill area, A_{fill}	= 196.523 m^2			
	Therefore, effective air velocity through fill, v_{fill}	$= G_{wm}/A_{fill}$			
		= 2.631 m/s			
	Water loading, WL	$= L/A_{fill}$			
		= 5.300 kg/sec/m^2			
	Therefore, fill pressure drop	= 12.753 mmWC			
c)	Pressure drop through spray zone	$= N_{vh\text{-}sz} \times \rho_{w2} \times (v_e)^2/2g$			
	$N_{vh\text{-}sz}$	$= sz \times (0.4 \times L/G + 1) \times \rho_{avf}/\rho_{avo} \times (m_{avo}/m_{avf})^2$			
		or $= sz \times (0.4 \times L/G + 1) \times \rho_{wm}/\rho_{w2} \times (m_{avo}/m_{avf})^2$			
		= 1.099			
	Assuming a column grid spacing of 4.7 m × 4.9 m in the rectangular cell size of 14.2 m × 14.7 m, the number of columns in a cell, C_N	= 9			
	Therefore, total column area per cell, A_c	= 2.3 m^2			
	Hence, effective air flow area in spray zone and plenum, A_e	$= CL \times CW - A_c$			
		= 206.817 m^2			
	Therefore, air velocity through spray zone and plenum, v_e	= 2.574 m/s			
	Hence, spray zone pressure drop	= 0.405 mmWC			
d)	Pressure drop because of the distribution piping system	$= N_{vh\text{-}DS} \times \rho_{w2} \times v_e^2/2g$			
		$= 0.775 \times \rho_{wm}/\rho_{w2} \times (m_{avo}/m_{avf})^2$			

Sl No.	Solved Example of an IDCT Thermic Design or Fresh Water Application				
	Therefore, pressure drop through distribution system	= 0.810 mmWC			
e)	The number of velocity heads for pressure drop through drift eliminators, plenum and fan entrance as given in the TP88 curve, $N_{vh\text{-}dpf}$	= 5			
	Therefore, pressure drop through drift eliminators, plenum and fan entrance	$= N_{vh\text{-}dpf} \times \rho_{w2} \times v_e^2/2g$			
		= 1.845 mmWC			
f)	Total static pressure, *SP*	= 18.115 mmWC			
10	Check for velocity pressure ratio, V_{pr}				
	One velocity head at air inlet, $1V_H$	= Air inlet pressure drop/$N_{vh\text{-}ai}$			
		= 0.768			
	Therefore, V_{pr}	$= SP/1V_H$			
		= 23.600	≥ 5		
11	Fan velocity pressure:				
a)	As the air-water loading is high because of the small space available for the IDCT in this solved example, the highest fan size possible in the chosen cell dimensions needs to selected to achieve the lowest power possible. However, this selection shall be based on client specifications that may or may not require some free space all around fan stacks.				
	Fan diameter, *D*	= 10 m			
	Therefore, fan area, A_d	= 78.540 m^2			
	Assume fan hub diametre, D_h	= 1.500 m	(Obtain actual hub diametre from manufacturer)		
	Therefore, hub area, A_h	= 1.767 m^2			
	Hence, effective fan airflow area, A_{ef}	= 76.773 m^2			
	Therefore, velocity across fan, v_{fan}	$= G_{w2}/A_{ef}$			

Sl No.	Solved Example of an IDCT Thermic Design or Fresh Water Application				
		= 6.934 m/s			
	Velocity pressure across fan, V_{P1}	$= \rho_{w2} \times v_{fan}^2/2g$			
		= 2.677 mmWC			
b)	Assume a venturi of 2.2 m height at an angle of 7.5 ºC. For best performance and to achieve the venturi efficiency of 70 percent stated in the guideline for a RCC IDCT				
	Stack exit diametre, D_{se}	$= D + TC + 2 \times 2.2 \times \tan(7.5)$			
	Tip clearance, TC	= 25 mm	(This may be achievable with FRP stacks but may not be possible in RCC, in which case fan ring may be used)		
	for RCC stack, say	= 40 mm			
	Therefore, D_{se}	= 10.619 m			
	Stack exit area, A_{se}	= 88.568 m^2			
	Therefore, stack exit velocity, v_{se}	$= G_{w2}/A_{se}$			
		= 6.010	≈ 6 m/s	(Venturi height may be adjusted to get exit air velocity ≥ 6 m/s)	
	Velocity pressure at stack exit, V_{P2}	$= \rho_{w2} \times v_{se}^2/2g$			
		= 2.0 mmWC			
c)	Velocity pressure recovery @ 70 percent venturi efficiency, V_r	$= (V_{P1} - V_{P2}) \times 0.7$			
		= 0.466 mmWC			
12	Fan total pressure, TP	$= SP + V_{P1} - V_r$			
		= 20.327 mmWC			
13	Fan power required, FKW	$= G_{w2} \times TP/(368\,000 \times \eta_{fan})$			
	η_{fan}	= 80 %			
	Therefore, FKW	= 132.31 kW			

Sl No.	Solved Example of an IDCT Thermic Design or Fresh Water Application				
14	At 95 percent gear box efficiency, the break shaft power, *BKW*	= 139.28 kW			
	Total BKW for all the operating cells	= 1 114.23 KW			
15	Check for heat balance				
	$(L \times C_p \times \text{Range}) + (E_v \times C_p \times T_2) = G_a \times \Delta h$				
	$E_v = (x_2 - x_1) \times G_a$				
	Therefore, *E*v	= 490 420.104 5 kg/h		(For all the operating cells)	
	The LHS of the equation will be	= 316 183 863.4 kcal/h			
	and The RHS of the equation will be	= 316 183 863.4 kcal/h			

ANNEX B

(*Foreword*)

SOLVED EXAMPLE OF AN IDCT THERMIC DESIGN FOR SEA WATER APPLICATION

Sl No.	Solved Example of an IDCT Thermic Design for Sea Water Application				
1	Design duty parameters:				
	Water flow rate, L	= 30 000 m^3/h			
	Hot water temperature, T_1	= 43 °C			
	Cold water temperature, T_2	= 33 °C			
	Wet bulb temperature, t_1	= 28 °C			
	Relative humidity RH_1	= 50 %			
	Salt content (in make-up water)	= 31 850			
	Elevation above main sea level, *EMSL*	= 0 m			
	Available pumping head above ground level, *TPH*	= 10.7 m (say)			
NOTE — Change in air properties may be considered in design only when EMSL is above 300 m (reference BS 4485/BSEN14705).					
2	Let's say that this sea water circulating system runs at a cycles of concentration (COC) of 1.5. This means that the salinity to be considered in thermic design will be (31 850 × 1.5) 47 775 ppm. The effect of this salinity in sea water is that it reduces the vapor pressure that in turn reduces the enthalpy,thus reducing the available driving force. As there are no enthalpy tables published in any code for sea water, the only way to design an IDCT for sea water application is by using equations for air properties. As mentioned in the previous example with fresh water, any set of published equations or tables for air properties from any code or standard can be used for thermic design to result in more or less identical IDCT sizing and fan power consumption. However, it shall be ensured that the data used and the method applied for calculations shall be uniform throughout to get proper results.				
	The thermic design method presented for sea water application is identical with that of fresh water except that equations are used for determining air properties for the saline condition. The vapor pressure is corrected for salinity using the equation given at **8.7.3** of BS 4485.				

Sl No.	Solved Example of an IDCT Thermic Design for Sea Water Application				
3	Fill selection for KaV/L calculation				
	Even though Marley Fill MC75 is not suitable for sea water application, same is used here to demonstrate how the L/G and resulting fan power varies with change in vapour pressure. The correct fill selection for sea water application should have been a splash fill.				
	Fill height, FH	= 1.8 m (say)			
	The performance equations of this fill presented at **3.4.3.4** of the guidelines are as under:				
	$(KaV/L)_{\text{fill}}$	$= 1.035 \times (L/G)^{-0.781} \times (3.28 \times FH)^{0.584}$			
	The air inlet is derived from the available pumping head as under:				
	Ground level to basin sill	= 0.3 m			
	Main beam depth, MBD	= 0.5 m (say)			
	Secondary beam depth, SBD	= 0.25 m (say)			
	Fill depth, FH	= 1.8 m or 6 feet			
	Spray height/zone, SZ	= 0.6 m (say)			
	Nozzle depth, ND	= 0.2 m (say)			
	Head required for nozzle	= 0.85 m (say)			
	Head loss in HW piping + distribution system	= 1.2 m (say)			
	Sub-total, HL	= 5.7			
	Possible air inlet height, $H_{1} = (TPH - HL)$	= 5 m			
	As the rain and spray zones are ignored, $(KaV/L)_{\text{A}}$	$= (KaV/L)_{\text{fill}}$			
	This $(KaV/L)_{\text{A}}$ will have to be matched with $(KaV/L)_{\text{D}}$ to determine the exit air temperature and the design L/G				
4	The heat balance equation of a cooling tower is $(L \times C_{\text{p}} \times \text{Range}) + (E_{\text{v}} \times C_{\text{p}} \times T_{2}) = G \times \Delta h$				
	where E_{v} = evaporation loss (kg/h)	E_{v} = loss can be calculated once inlet and outlet air properties are established			

Sl No.	Solved Example of an IDCT Thermic Design for Sea Water Application				
	Specific heat, C_p	= 1 kcal/kg/ºC			
	Δh = inlet enthalpy – outlet enthalpy	= ($h_1 - h_2$) kcal/kg			
	Range = ($T_1 - T_2$)	= 10 ºC			
	G = Dry air flow rate (kg/h)				
	Solving this equation without knowing what the L/G and exit air temperature would be is a little cumbersome and requires an iterative process. The CTI ATC-105 heat balance equation ignores the heat of evaporated water for the sake of simplifying the calculations. However, in actual practice where design performance is guaranteed, this heat from the evaporated water cannot be ignored and hence, iterative calculations are required to be carried out. Equations are readily available for the calculation of air properties in BS, ASHRAE, DIN, Kroger, etc publications. However, air properties using equations in Kroger's book are used here for demonstration purposes as manually readings values from tables is extremely laborious. Though the CTI tables can be used for IDCT design assuming that the inlet *RH* is 100 percent, air properties at the inlet relative humidity will be insisted upon by clients in data sheets. It may be noted that the $(KaV/L)_D$ value varies with the tables/equations from different publications because of the datum differences. However, a uniform method of calculation based onany of these publications in their entirety will result in more or less comparable IDCT sizing and fan power consumption.				
5	Inlet air properties (from equations):				
	Design inlet WBT, t_1	= 28 °C			
	Relative humidity, RH_1	= 50 %			
	Dry bulb temprature (*DBT*), t_{a1}	= 37.14 °C			
	Wet-air density, ρ_{w1}	= 1.1239 kg/m^3			
	Absolute humidity, x_1	= 0.0202 kg/kg			
	Enthalpy, h_1	= 21.366 kcal/kg			
6	For a conservative design, it shall be assumed that the exit air is fully saturated, that is, its *RH* is 100 percent				
	Now, assume an exit air temperature of say, $[(T_1 + T_2)/2]$ = (43 + 33)/2 = 38 °C for the first iteration				
	Exit air properties (from equations):				
	WBT (t_2) @ 100 percent *RH*	= 38 °C			

Sl No.	**Solved Example of an IDCT Thermic Design for Sea Water Application**				
	Relative humidity, RH_2	= 100 %			
	Dry bulb temprature (*DBT*), t_{a2}	= 38 °C			
	Abs humidity, x_2	= 0.043 8 kg/kg			
	Enthalpy, h_2	= 36.045 kcal/kg			
	Now rearrange the heat balance equation for L/G as under:				
	$L/G = (\Delta h - (x_2 - x_1) \times C_p \times T_2)/(C_p \times \text{Range})$				
	Substituting values, we get L/G	= 1.390			
	Now, as the L/G and exit air enthalpy are both known, the $(KaV/L)_D$ can be calculated using the Tchebycheff method				
	(Refer ATC-105 or BS 4485 codes for a detailed description of the Tchebycheff method)				
	As per the Tchebycheff method all enthalpies are considered @ 100 percent *RH*.				
Iteration No.1	**Water Temperature Distribution (between T_1 & T_2)**	**Enthalpy at Water Temperature (h_w)**	**Enthalpy Distribution (h_a)**	**(h_w - h_a)**	**1/(h_w - h_a)**
	33		21.366		
	34	29.077	22.757	6.3204	0.15822
	37	33.857	26.927	6.9300	0.14430
	39	37.442	29.708	7.7340	0.12930
	42	43.509	33.878	9.6309	0.10383
	43		35.269		
				$\Sigma 1/(h_w - h_a)$	0.535 65
	As per Tchebycheff method $(KaV/L)_D$ = Range/4 × $\Sigma 1/(h_w - h_a) \times C_p$ Therefore, $(KaV/L)_D$			= 1.339	
	Now, let's check how well $(KaV/L)_A$ meets the $(KaV/L)_D$ calculated				
	We have $(KaV/L)_A = 1.035 \times (L/G)^{-0.781} \times (3.28 \times FH)^{0.584}$				
	Substituting L/G value, we have $(KaV/L)_A$	= 2.257 1			
Iteration No. 2	As the $(KaV/L)_D$ and $(KaV/L)_A$ do not match, the exit air temperature will have to be iterated until a close match is achieved				

Sl No.	Solved Example of an IDCT Thermic Design for Sea Water Application				
	Exit Air Properties (from Equations)				
	Wet bulb temprature (*WBT*), t_2 @ 100 percent *RH*	= 40.35 °C			
	Relative humidity, RH_2	= 100 %			
	Dry bulb temprature (*DBT*), t_{a2}	= 40.35 °C			
	Wet-air density, ρ_{w2}	= 1.094 1 kg/m^3			
	Abs humidity, x_2	= 0.050 1 kg/kg			
	Enthalpy, h_2	= 40.570 kcal/kg			
	Substituting values, we get *L/G*	= 1.822			
	Water Temperature Distribution (between T_1 and T_2)	**Enthalpy at Water Temperature (h_w)**	**Enthalpy Distribution (h_a)**	**($h_w - h_a$)**	**1/($h_w - h_a$)**
	33		21.366		
	34	29.077	23.188	5.888 8	0.169 81
	37	33.857	28.653	5.204 0	0.192 16
	39	37.442	32.297	5.144 9	0.194 37
	42	43.509	37.762	5.747 2	0.174 00
	43		39.584		
				$\Sigma 1/(h_w - h_a)$	0.730 34
	As per Tchebycheff method $(KaV/L)_D$ = Range/4 × $\Sigma 1/(h_w - h_a) \times C_p$ Therefore, $(KaV/L)_D$			= 1.826	
	Now, let's check how well $(KaV/L)_A$ meets the $(KaV/L)_D$ calculated				
	Substituting *L/G* value, we have $(KaV/L)_A$	= 1.827			
	The iterations will end here as $(KaV/L)_A$ meets the $(KaV/L)_D$ calculated at the assumed exit air temperature and the air properties established in this iteration will be used for further calculations.				

Sl No.	Solved Example of an IDCT Thermic Design for Sea Water Application				
7	The rest of the calculations are simple as there are no more iterations to be performed. The number of cells and individual cell dimensions depend on the available space for the IDCT and the optimization of fan power consumption and/or overall cost of the IDCT will depend on evaluation criteria specified in tenders that vary with client. For this solved example, let's assume that the maximum space available is 130 m (L) × 15 m (W) excluding space required for staircases, CW channel, etc and that the minimum number of cells to be provided is eight working plus one standby.				
	As the minimum number of operating cells required is eight, it is possible to design the IDCT with more number of cells with back-to-back orientation. Optimization between back-to-back and single row cell arrangement will require lot many design runs and application of loading factors (on power and initial cost), which users of this guideline may carryout independently. Let's assume that the single row option with eight working plus one standby cell is the most optimal for this solved example.				
	Total available tower length	= 130 m			
	Assuming 500 mm wide columns in air inlet and 150 mm thick cell partition walls, space occupied by walls and end columns	= 2 m (approx)			
	Actual length available of IDCT design	= 128 m			
	Total number of cells	= 9			
	Cell length, *CL*	= 14.2 m (say)			
	Similarly, actual available cell width, *CW*	= 14.7 m (say)			
8	Water flow per operating cell	= 3 750 m^3/h			
	Dry air flow per cell, G_a	= $L/(L/G)$			
	Therefore, G_a	= 2 058 468.911 kg/h			
		= 571.797 kg/s			
	Wet airflow through inlet G_{w1}	= $G_a \times (1 + x_1)/\rho_{w1}$			
		= 519.033 m^3/s			
	Average wet air density, $\rho_{wm} = (\rho_{w1} + \rho_{w2})/2$	= 1.109 043 758			
	Absolute humidity in fill zone, $x_m = (x_1 + x_2)/2$	= 0.035 16			

Sl No.	Solved Example of an IDCT Thermic Design for Sea Water Application				
	Average wet air flow through fill, $G_{wm} = G_a \times (1 + x_m)/\rho_{wm}$	= 1 921 342.343 m^3/h			
		= 533.71 m^3/s			
	Average wet air mass flow through fill, m_{avf}	= 591.90 kg/s			
	Wet air flow above fill or through exit, G_{w2}	$= G_a \times (1 + x_2)/\rho_{w2}$			
	Therefore, G_{w2}	= 548.78 m^3/s			
	Wet air mass flow above fill or through exit, m_{avo}	= 600.44 kg/s			
9	We can now establish the pressure drops in various zones of the IDCT using equations given **3.4.3** in the guideline				
a)	Air inlet area, A_{ai}	= (CL – end column width) × H_1 × 2 sides			
		= 137.222 m^2			
	Air velocity through inlet, v_{ai}	$= G_{w1}/A_{ai}$			
		= 3.782 m/s			
	Number of velocity heads in air inlet as per TP - 88 curve, $N_{vh\text{-}ai}$	= 3			
	Therefore, air inlet pressure drop	$= N_{vh\text{-}ai} \times \rho_{w1} \times (v_{ai})^2/2g$			
		= 2.459 mmWC			
b)	Fill pressure drop equation	$= (0.000\ 000\ 043\ 4 \times (196.8\times v_{fill})^{2.355\ 9} + 1.472\ 34 \times WL \times 0.000\ 000\ 811 \times (196.8\times v_{fill})^{1.540\ 3} \times (1 + 0.283 \times 3.28\times FH) \times \rho_{wm}/1.2$			
	Gross fill area	$= CL \times CW$			
		= 209.067 m^2			

Sl No.	Solved Example of an IDCT Thermic Design for Sea Water Application				
	There will be no fill in the cell wherever columns exist. Assume 6 percent column and beam obstructions for this solved example, meaningthe effective fill area, A_{fill}	= 196.523 m^2			
	Therefore, effective air velocity through fill, v_{fill}	= G_{wm}/A_{fill}			
		= 2.716 m/s			
	Water loading , *WL*	= L/A_{fill}			
		= 5.300 kg/sec/m^2			
	Therefore, fill pressure drop	= 13.594 mmWC			
c)	Pressure drop through spray zone	= $N_{vh\text{-}sz} \times \rho_{w2} \times (v_e)^2/2g$			
	$N_{vh\text{-}sz}$	= sz × (0.4 × L/G + 1) × ρ_{avf}/ρ_{avo} × $(m_{avo}/m_{avf})^2$			
		or = sz × (0.4 × L/G + 1) × ρ_{wm}/ρ_{w2} × $(m_{avo}/m_{avf})^2$			
		= 1.082			
	Assuming a column grid spacing of 4.7 m × 4.9 m in the rectangular cell size of 14.2 m × 14.7m, the number of columns in a cell, C_N	= 9			
	Therefore, total column area per cell, A_c	= 2.3 m^2			
	Hence, effective air flow area in spray zone and plenum, A_e	= $CL \times CW - A_c$			
		= 206.817 m^2			
	Therefore, air velocity through spray zone & plenum, v_e	= 2.653 m/s			
	Hence, spray zone pressure drop	= 0.425 mmWC			
d)	Pressure drop because of the distribution piping system	= $N_{vh\text{-}ds} \times \rho_{w2} \times v_e^2/2g$			
		= 0.775 × ρ_{wm}/ρ_{w2} × $(m_{avo}/m_{avf})^2$			
	Therefore, Pressure drop through Distribution System	= 0.808 mmWC			

Sl No.	Solved Example of an IDCT Thermic Design for Sea Water Application				
e)	The number of velocity heads for pressure drop through drift eliminators, plenum andfan entrance as given in the TP88 curve, $N_{vh\text{-}dpf}$	= 5			
	Therefore, pressure drop through drift eliminators, plenum and fan entrance	$= N_{vh\text{-}dpf} \times \rho_{w2} \times v_e^2/2g$			
		= 1.963 mmWC			
f)	Total static pressure, SP	= 19.249 mmWC			
10	Check for velocity pressure ratio, V_{pr}				
	One velocity head at air inlet, $1V_H$	= Air inlet pressure drop/$N_{vh\text{-}ai}$			
		= 0.820			
	Therefore, V_{pr}	$= SP/1V_H$			
		= 23.487	≥ 5		
11	Fan velocity pressure				
a)	As the air-water loading is high because of the small space available for the IDCT in this solved example, the highest fan size possible in the chosen cell dimensions needs to be selected to achieve the lowest power possible. However, this selection shall be based on client specifications that may or may not require some free space all around fan stacks.				
	Fan diametre, D	= 10 m			
	Therefore, fan area, A_d	= 78.540 m^2			
	Assume fan hub diametre, D_h	= 1.500 m	Obtain actual hub diametre from manufacturer		
	Therefore, hub area, A_h	= 1.767 m^2			
	Hence, effective fan airflow area, A_{ef}	= 76.773 m^2			
	Therefore, velocity across fan, v_{fan}	$= G_{w2}/A_{ef}$			
		= 7.148 m/s			
	Velocity pressure across fan, V_{P1}	$= \rho_{w2} \times v_{fan}^2/2g$			
		= 2.849 mmWC			

Sl No.	Solved Example of an IDCT Thermic Design for Sea Water Application				
b)	Assume a venturi of 2.8 m height at an angle of 7.5 ºC. For best performance and to achieve the venturi efficiency of 70 percent stated in the guideline				
	Stack exit diametre, D_{se}	$= D + TC + 2 \times 2.8 \times \tan(7.5)$			
	Tip clearance, TC	= 25 mm		(This may be achievable with FRP stacks but may not be possible in RCC, in which case fan ring may be used)	
	for RCC stack, say	= 40 mm			
	Therefore, D_{se}	= 10.777 m			
	Stack exit area, A_{se}	= 91.223 m^2			
	Therefore, stack exit velocity, v_{se}	$= G_{w2}/A_{se}$			
		= 6.016 m/s	≥ 6 m/s	(Venturi height may be adjusted to get exit air velocity ≈ 6 m/s)	
	Velocity pressure at stack exit, V_{P2}	$= \rho_{w2} \times v_{se}^2/2g$			
		= 2.0 mmWC			
c)	Velocity recovery @ 70 percent venturi efficiency, V_r	$= (V_{P1} - V_{P2}) \times 0.7$			
		= 0.582 mmWC			
12	Fan total pressure, TP	$= SP + V_{P1} - V_r$			
		= 21.517 mmWC			
13	Fan power required, FKW	$= G_{w2} \times TP$ / (368 000 × η_{fan})			
	η_{fan}	= 80 %			
	Therefore, FKW	= 144.39 Kw			
14	At 95 percent gear box efficiency, the break shaft power, BKW	= 151.99 kW			
	Total BKW for all the operating cells	= 1 215.94 kW			

Sl No.	Solved Example of an IDCT Thermic Design for Sea Water Application				
15	Check for heat balance:				
	$L \times C_p \times \text{Range} + E_v \times C_p \times T_2 = G_a \times \Delta h$				
	$E_v = (x_2 - x_1) \times G_a$				
	Therefore, E_v	= 491 932.824 9 kg/h		(For all the operating cells)	
	The LHS of the equation will be	= 316 233 783.2 kcal/h			
	The RHS of the equation will be	= 316 233 783.2 kcal/h			

ANNEX C

(*Clause* 5.4.5)

THERMAL AND PRESSURE DROP CALCULATIONS

C-1 This has already been done by various cooling tower designers and many reports and technical bulletins are available for reference. It is advised that the recommendations in the subsequent sections be followed as a minimum to arrive at a reasonably accurate thermal design of IDCTs. However, the designer may improve on the recommended pressure drops based on their own research information to render the designs accurate and conservative.

C-2 Air inlet pressure drop is one of the significant factors that affect the overall sizing and fan power requirement of an IDCT. It is recommended that this pressure drop be calculated from the graph given at Fig. 20.

NOTE — Figure 20 is taken from TP88-05 published by CTI.

C-3 Rain zone in an IDCT is small compared to that in a NDCT.

NOTE — TP-88 does not consider the *KaV/L* gain in the rain zone and hence, this guideline recommends the same. However, if any end user/tender specifications permit the same, designers may consider this gain along with additional pressure drops, if any in their thermal design. Refer C-9 for a detailed explanation regarding end effects in fill *KaV/L*.

C-4 The pressure drop in the air path through the rain zone is assumed to be included in the '3-velocity heads' curve reproduced above from TP88-05. Additional pressure drops, if any due to the column-beam structure supporting the fill may be considered in thermal designs.

C-5 The fill performance characteristics shall be taken from the manufacturer's published data. However, such performance characteristic data for some of the established fills that are commonly used in the industry is given below.

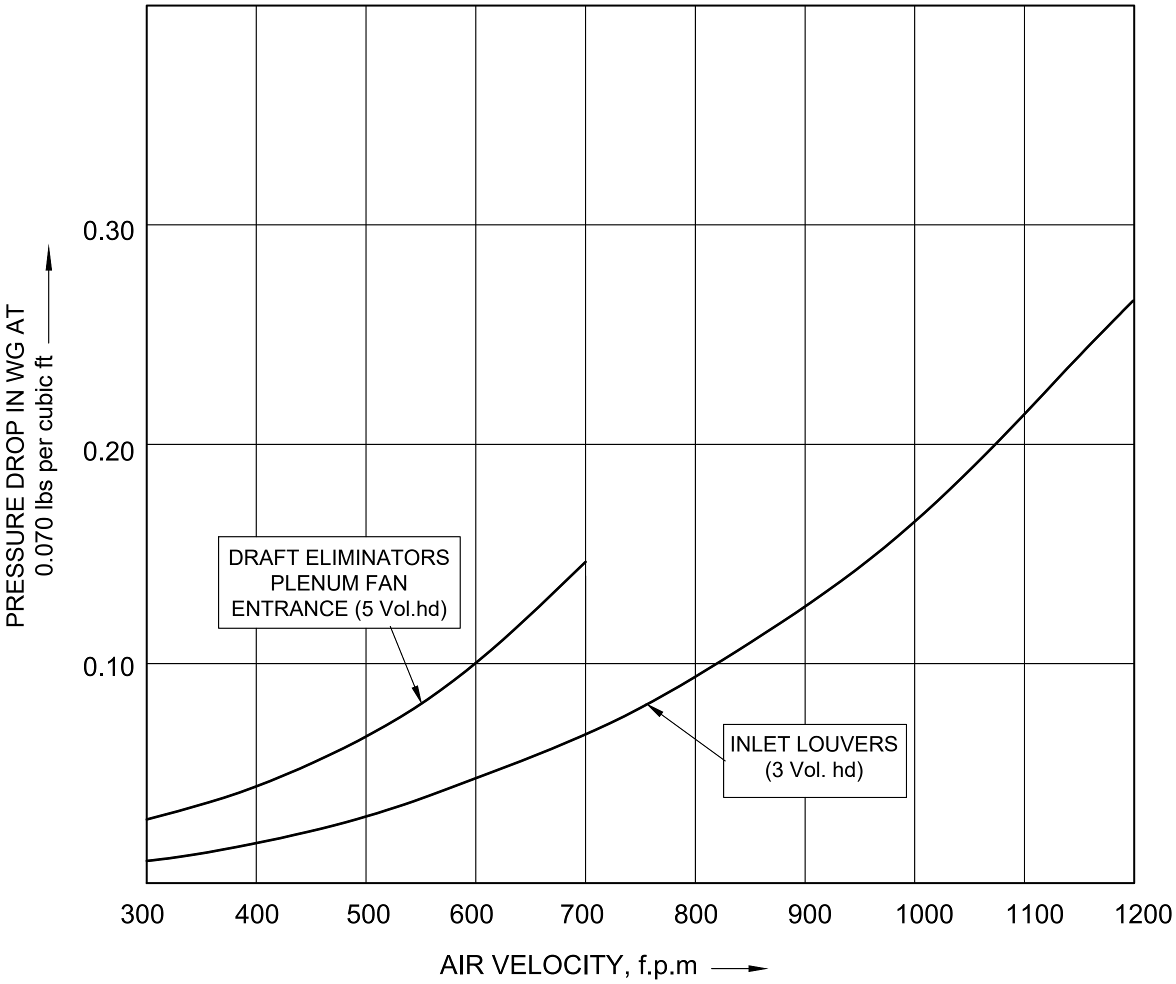


FIG. 20 IDCT PRESSURE DROPS

C-6 The fill performance data presented in this guideline is taken from data provided by manufacturers and that available in public domain. All the data presented is for face velocities between 300 and 700 FPM. However, the manufacturers and research bodies like EPRI have advised that the fill performance for face velocities below 300 FPM can be determined using the same fill equations with accuracy as the performance is enhanced with reducing face velocities. Hence, the pressure drop equations for some of the fills used in the industry that were tested for face velocities up to 700 FPM are as under:

a) 19 mm cross fluted film fill,

$$KaV/L = 1.864 \times (L/G)^{-0.862\,1} \times (3.28 \times FH)^{0.876\,4} \times (196.8 \times v_{\text{filll}})^{-0.190\,2}$$

$$\Delta p = \left(C_0 + C_1 \times WL \times 1.472\,34 + C_2 \times v_{\text{fill}} \times 196.8 + C_3 \times v_{\text{fill}} \times 196.8 \times WL \times 1.472\,34 + C_4 \times (1.472\,34 \times WL)^2 + C_5 \times (196.8 \times v_{\text{fill}})^2 + C_6 \times 196.8 \times v_{\text{fill}} \times (1.472\,34 \times WL)^2 + C_7 \times \left(196.8 \times v_{fill}\right)^2 \times 1.472\,34 \times WL\right) \times (FH \times 3.28)^{0.75} \times \ (\rho_{avf}\ /1.12) \times 25.4 \text{ (mmWC)}$$

where

C_0 = -0.000 23;
C_1 = 0.001 915 7;
C_2 = 4.177 1 × 10^{-5};
C_3 = -1.119 7 × 10^{-5};
C_4 = -4.342 2 × 10^{-5};
C_5 = 1.825 8 × 10^{-7};
C_6 = 4.573 9 × 10^{-7}; and
C_7 = 1.93 × 10^{-8}.

NOTE — Data has been taken from Munters.

b) 27 mm cross fluted film fill (C.10.27 of Munters)

$$KaV/L = 0.57 \times (L/G)^{-0.722\,7} \times (3.28 \times FH)^{0.670\,6} \times (196.8 \times v_{\text{fill}})^{-0.027\,45}$$

$$\Delta p = (1.472\,34 \times WL \times (C_{x0} \times 196.8 \times v_{\text{fill}} - C_{x1}) + C_{x2} \times (196.8 \times v_{\text{fill}})^2) \times (FH \times 3.28)^{0.7} \times (\rho_{avf}/1.12) \times 25.4 \text{ (mmWC)}$$

where

C_{X0} = 9.513 5 × 10^{-6};
C_{X1} = 0.000 758 3; and
C_{X2} = 1.648 × 10^{-7}.

NOTE — Data has been taken from Munters.

c) MC 75 cross fluted packing.

$$KaV/L = 1.035 \times (L/G)^{-0.781} \times (3.28 \times FH)^{0.584}$$

$$\Delta p = \left(4.34 \times 10^{-8} \times (196.8 \times v_{\text{fill}})^{2.355\,9} + 1.472\,34 \times WL \times 8.11 \times 10^{-07} \times 196.8 \times (v_{fill})^{1.540\,3}\right) \times (1 + 2.83 \times \ 10^{-01} \times 3.28 \times FH) \times \rho_{\text{avf}}/1.2 \times 25.4. \text{ (mmWC)}$$

NOTE — Data has been taken from Marley.

d) Splash grid/PP grid — 200 mm spacing (Cooldeck)

$$KaV/L = 0.71 \times (L/G)^{-0.42} \times (FH)^{0.5}$$
$$\Delta p = 2.88 \times WL^{0.85} \times (WL/(L/G))^{-0.6}$$

$$\Delta p = 2.88\ \times WL^{0.85} \times (WL/(L/G))^{-0.6} \times FH^{1.17} \times \rho_{\text{avf}}/\ (v_{\text{fill}})^2/2g \qquad \text{(mmWC)}$$

NOTE —This equation for splash grid/PP grid is taken from Dr. Kruger’s book. All equations presented above include end effects.

C-7 Users are advised to consult the respective manufacturers of the above types of fills and get the equations reaffirmed before using them in their calculations and/or computer programmes. Requirements defined in C-8 below may also be considered by designers before using the above equations in their thermal designs.

C-8 It has become a practice now a days for end users to specify that the fill being proposed to be used by the contractor in their project be tested in a third-party test facility to evaluate its thermal performance and that the data acquired from the test rig be used for thermal design. Hence, the fill performance data presented in this guideline must be carefully chosen by users based on manufacturer feedback, their own experience and research and specific tender conditions that require performance guarantees.

C-9 Spray Zone

The spray zone is the gap between the top of fill surface and the bottom of the spray nozzle. The minimum number of velocity heads for the spray zone should be calculated using the following formula:

$$N_{\mathrm{VH-SZ}} = \mathrm{sz} \times (0.4 \times L/G + 1) \times \rho_{avf}/\rho_{avo} \times \left(m_{\mathrm{avo}}/m_{avf}\right)^2 \text{(in velocity heads)}$$

Even though a small KaV/L gain occurs in the spray zone when the tower is new, it is soon lost within the first 2 to 3 years as fouling of the system begins. However, if the project specific tender specifications have no restrictions on this aspect and permits this gain in the spray zone, designers may consider the same in their thermal designs, provided they have nozzle specific performance data in terms of droplet size distribution and the associated *KaV/L* contribution established in a test rig for IDCTs.

In case fill test data for *KaV/L* is presented by the manufacturer (in-house or third-party test) to include end effects, then such resultant *KaV/L* equations may be used for thermal design, in which case no additional *KaV/L* equations be used for spray and rain zones as they are already built into the fill *KaV/L* equations. And in case the test data presented excludes the end effects in fill *KaV/L*, then project specifications should govern regarding consideration of additional *KaV/L* from spray zone and rain zone.

C-10 The distribution system inside the IDCT is a network of lateral pipes carrying numerous nozzles at specified locations. This piping system is a small obstruction to the air flow and hence, the minimum number of velocity heads for the distribution system should be calculated using the following formula:

$$N_{\mathrm{VH-DS}} = 0.775 \times \rho_{avf}/\rho_{avo} \times \left(m_{\mathrm{avo}}/m_{avf}\right)^2$$
(in velocity heads)

C-11 The pressure drop through drift eliminator, together with the turning losses in plenum and fan entering losses should be determined from the graph presented at **C-2**.

C-12 The velocity pressure across the fan should be calculated using the formula:

$$\rho_{avo} \times (Fan\ Airflow/(\pi\,(D^2 - D_{\mathrm{h}}^2)/4))^2/2g \text{ (in mmWC)}$$

C-13 Depending on whether a velocity recovery stack has been adopted or not, the velocity pressure across the fan should be calculated using the formula:

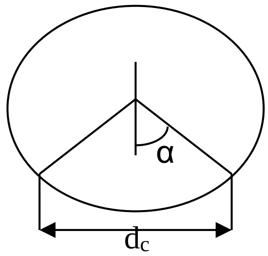
$$\rho_{avo} \times (Fan\ Airflow/(\pi \times D_{\mathrm{se}}/4))^2/2g$$
(in mmWC)

C-14 RCC fan Stacks seldom achieve a smooth aerodynamic profile and close tip clearances unlike those in FRP construction. Hence, the velocity recovery efficiency to be used for a RCC fan stack should not exceed 70 percent. Similarly, that for a FRP fan stack should not exceed 75 percent, provided the recovery cone angle is between 7° and 8° and not more.

ANNEX D

(*Clause* 5.1.6.3)

SOLVED EXAMPLE FOR DETERMINING PVC DISTRIBUTION PIPE SPAN/DEFLECTION

Pipe diameter (OD = 2r)	:	0.160	m		
Pressure rating, say Class 3 as per IS 4985	:	6.0	kg/cm^2	4	
Thickness of pipe (min) as per IS 4985	:	6.3	mm		
Supporting span (assumed)	:	1.0	m		
Specific gravity	:	14.22	KN/m^3		
Modulus of elasticity u-PVC at 27 °C	:	1 820	Mpa	(this value varies from 1.82 to 3 Gpa; the lowest value is considered for conservativeness)	
Permissible bending stress at 27 °C	:	5.52	Mpa		
Deration factor for 50 °C	:	0.6		(As per Fig. 1 of IS 4985)	
Permissible bending stress at 50 °C	:	3.312	Mpa	(= 5.52 × 0.6)	
Modulus of elasticity	:	1 092	Mpa	(= 0.6 × 1820)	
Calculation of moment of inertia of pipe with hole (for fixing nozzle adapter):					
Pipe ID	:	0.147 4	m		
I	:	8.998 × 10^{-06}	m^4	$= \pi/64\ (D^4 - d^4)$	
As the PVC pipe is punctured to fix adapters for nozzles, this loss of solidity is to be considered for calculation of moment of inertia					
Hole dia in pipe	:	0.063	m		
Half angle subtended by hole at centre of pipe, α	:	0.404 7	radians	$= \sin^{-1}(0.063/2/0.08)$	
		23.19	degrees		
A_c	:	3.919 × 10^{-04}	m^2	$= \alpha/4(d_o^2 - d_i^2)$	
I_c	:	2.376 × 10^{-06}	m^4	$I_c = \frac{A_c \times r^2}{2}\left(1 + \frac{\sin\alpha \times \cos\alpha}{\alpha}\right)$	
I_e	:	6.623 × 10^{-06}	m^4	$= I - I_c$	

Z	:	8.278×10^{-05}	m^3	$= I_e / r$	
Weight of pipe	:	43.27	N/m		
Water load		167.4	N/m		
Miscellaneous load	:	784.8	N/m		
Total load, W		995.5	N/m		
Check for Bending Stress:					
Bending moment, M	:	124.43	N-m	$= WL^2/8$	
Bending stress	:	1.503×10^6	Pascals	= M/Z	
		1.50	Mpa		
		SAFE (Less than permissible 3.312 Mpa)			
Check for Deflection:					
Maximum deflection, δ	:	0.001 8	m	$= 5WL^4/ (384 \times E \times I_e)$	
L/δ (span to deflection ratio)	:	557.9			
		> = 500 SAFE			

ANNEX E

(*Foreword*)

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