

### Radiator & Oil Cooler Thermal Design Parameters for Applications other than Passenger Vehicles.

1. Passenger vehicles very rarely operate under full load conditions, and being petrol powered, very few with lube oil cooling - operate at very high water temperatures and water pressures, and have an abundance of free ram air. Therefore, design philosophy follows a different path to heavy equipment.

The non-passenger vehicle - heavy truck, bus, industrial, stationary, locomotive, earth-moving, agricultural, power generating equipment, is primarily diesel powered with maximum load characteristics, controlled maximum allowable water temperatures and generally no "free air".

So we have two totally contrasting conditions to which radiator design parameters are applied.

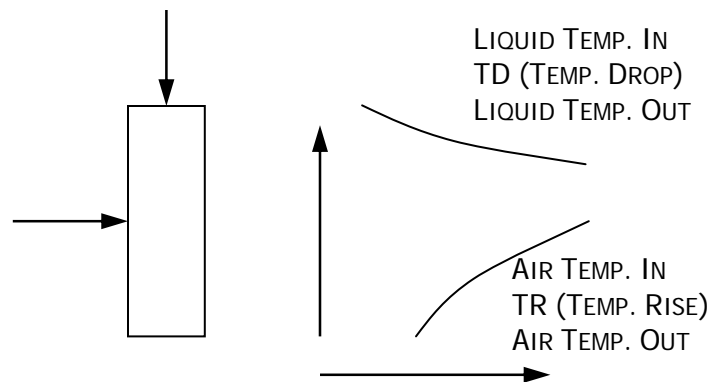
There have been numerous applications where attempts have been made to use automotive radiators for stationary industrial purposes with disastrous results - primarily over-heating.

This discussion is based on radiator thermal design parameters for the non-passenger vehicle area of equipment.

2. In the broad sense, heat dissipation of a radiator core is a function of mass flows, temperature differentials of the liquid to be cooled and the cooling air, and the heat transfer surface used.

The water temperature being stabilised at a balance point of the above.

For a given amount of heat, the greater the temperature difference, between water to air, the lesser the surface area required, and vice versa.



3. As we are more concerned with the radiator design parameters, and the installation, we do not propose to delve too deeply into the theory of thermal conductivities of various cores - we believe the discussion should relate to the practical aspects associated with the input design information and its effects in an installation.

4. Selection of Core:

In the selection of a radiator core for any application, there are a basic set of design parameters that the designer needs to know.

These are:-

1. Air temperature at point of entry to core
2. Heat rejection
3. Flow rate of liquid to be cooled and available pressure
4. Maximum allowable temperature of liquid
5. Amount of cooling air available
6. Physical limitation

As each engine / application varies in heat, flow, maximum temperatures, fans, fan ratios - the above factors have to be reduced to a common factor by calculation and related to known core performance curves.

5. To arrive at this common factor, there are two methods used :-  
Arithmetic M.T.D. or Log M.T.D.

The arithmetic method does not take into account air rise over the core and can be used where M.T.D. of 60°F or greater is involved.

Where the M.T.D. is less than 60°F log M.T.D. is recommended as you will see shortly.

As Log. is more complex, we will restrict our discussion to arithmetic.

Temperature Rise (drop) is a function of:-

$$\text{Temperature Rise} = \frac{\text{Heat}}{\text{Mass Flow} \times \text{Sp Ht}}$$

Unless otherwise stated.

Where for our purposes in imperial terms we use :-

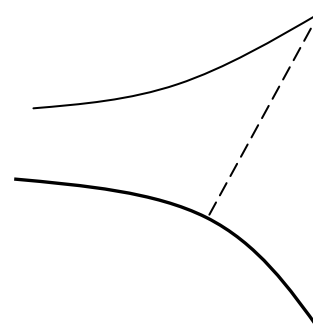
<u>Water</u>	<u>Sp Wt</u>	<u>Sp Ht</u>
	10 lbs / Imp Gall at 30°C (85°F)	1
	9.2 lbs / Imp Gal at 93°C (200°F)	
<u>Oil</u>	7-9 lbs / Imp Gal (8.5)	0.4-0.7 (0.46)
<u>Air</u>	0.069 lbs Wt/C.F At 45°C (113°F)	0.238

For Water

$$\begin{aligned} \text{Therefore, temperature drop (rise)} \\ = \frac{\text{Heat}}{\text{IMP GPM} \times 10 \times 1} \end{aligned}$$

$$\text{For Oil} = \frac{\text{Heat}}{\text{IMP GPM} \times 8.5 \times 0.46}$$

$$\text{For Air} = \frac{\text{Heat}}{\text{CFM} \times 0.069 \times 0.238}$$



Having established the temperature drop and approach air. We must tie those together.

Mean Temperature Difference

Is the Inlet Water Temperature - minus half the Temperature Drop - minus the Approach Air Temperature.

M.T.D. = Max. Water Temperature - 1/2  
T.D. - Ent. Air.

"U" Factor - is the term used for heat dissipation of any core - the core being rated at BTU/MIN/°F/sq. ft. M.T.D.

To achieve this :-

$$U = \frac{\text{Heat}}{\text{M.T.D.} \times \text{Area available}}$$

There are variables to be considered where change of temperatures are concerned.

- Water is not a Constant Density
- Air is not a constant mass
- Core performance varies with water flow and density

Let's take an example, and to keep it simple, we will exclude the variables.

Ambient (Approach Air) 100°F  
Heat to be rejected 10,000 BTU/MIN  
Water Flow 100 GPM (Imp)  
Max. Water 200°F  
Available Area (Core) 36" x 36" (9 sq. ft.)

$$T.D. = \frac{10,000}{100 \times 10} = 10^\circ\text{F}$$

$$M.T.D. = 200 - 5 - 100 = 95$$

$$U = \frac{10,000}{95 \times 9 \text{ sq. ft.}} = 11.69 \text{ BTU/MIN/}^\circ\text{F/sq. ft.}$$

Explain :- As can be seen

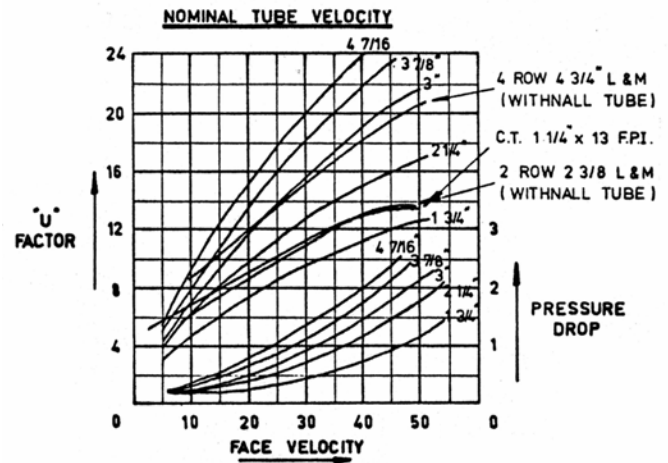
- Increase the heat - a higher "U"
- Increase the ambient - a higher "U"
- Decrease the area - a higher "U"

Core performance varies with water flow.

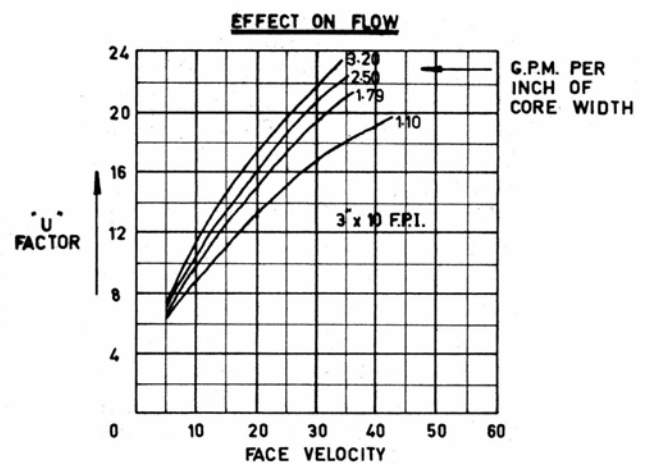
We have a "U" of 11.69 BTU/MIN/°F/sq. ft. M.T.D.

Lets look at Typical Core Performance.

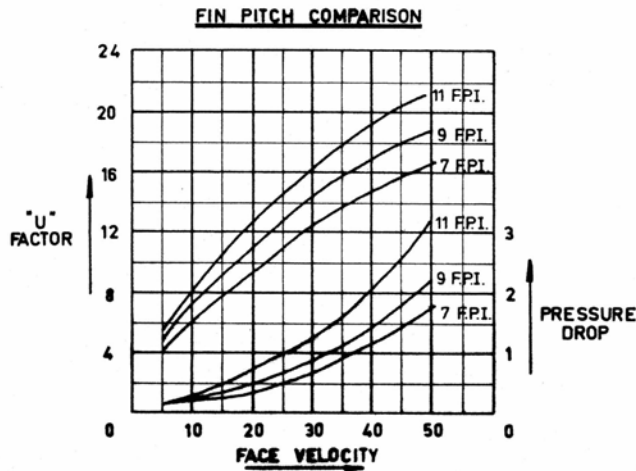
**Chart 'A'**



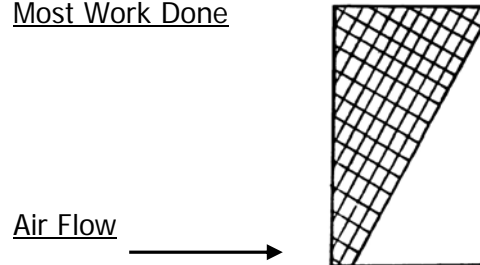
**Chart 'B'**



**Chart 'C'**



Most Work Done



Therefore, the thinnest core gives best value in terms of dollars for work done.

- There are numerous types of cores. The charts are related to fin and tube (F & T) cores as these have the balance of durability, "U", and pressure drop.

The C.T. core is primarily used on automotive, however a well constructed unit can be engineered into known applications.

Its performance:  
1.25" x 12 FPI is approximately equivalent to a 2.25" x 11 FPI F&T core.

The withnall:  
L&M 4 row is approximately equivalent to a 3" x 9 FPI F&T core

Therefore, each core has its own "U" characteristics and P.D.'s.

Returning to the chart, it can be seen that the "U" factor does not increase proportionately with the increase in thickness, this is a result of air temperature rise.

- Chart B Shows the effect on water flow. However physical size, fan location, etc., often prohibit the opportunity to take full advantage of this factor.

Chart C Shows the effect on varying fin pitch and the associated air pressure drops.

- The application invariably dictates the pitch.

eg:	Automotive	11 - 12.5 FPI
	Industrial	9 FPI
	Agricultural	5 - 7 FPI

The mechanical aspects of the cores are covered later.

- Now returning to our sample calculation :-

Our "U" was 11.69

Our area was 9 sq. ft.

Lets say we required a 3" x 9" FPI core and the "U" was achieved at 20 ft/sec and the air P.D. was 0.7" W.G.

The air flow required to meet our condition would be :-

$$9 \text{ sq. ft.} \times 20 \text{ ft/s} \times 60 \text{ sec} = 10800 \text{ CFM}$$

Our resistance to air flow            0.7" SWG  
 Plus our static head resistance    0.1" SWG  
  
 Total                                        0.8" SWG

Now having established our core size and air flow required, in perusing the core curves you will note that the "U" factor increases with air velocity.

Depending on the core type, a variation of 2 ft/sec can effect the water temperature by 3-5°F. This of course is a general comment.

To achieve the air flow required we require a fan.

A fan properly installed together with a cowl of suitable design for the application.

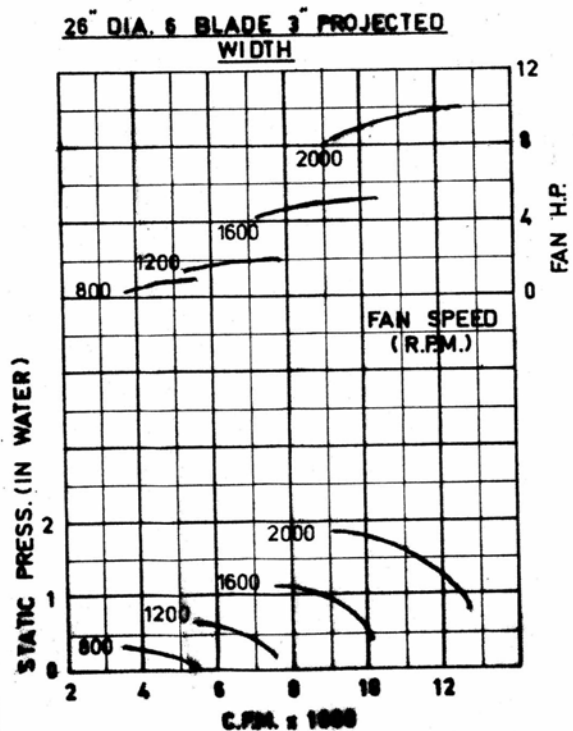
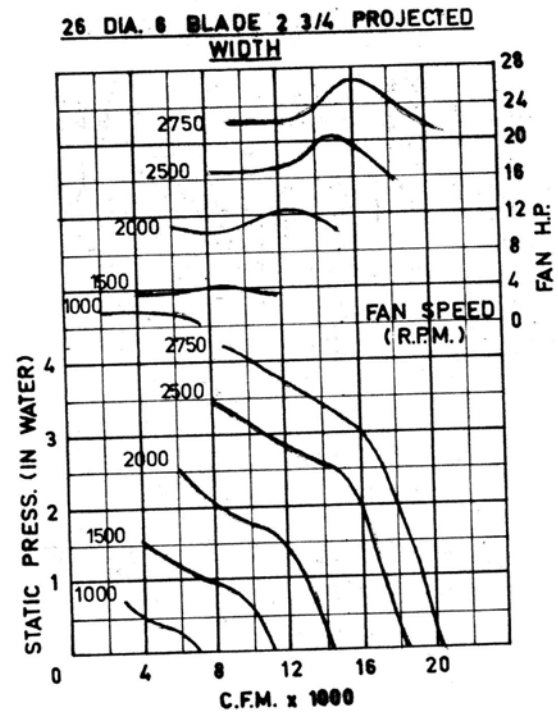
10. There are fans and fans - compare the performance of these two

The typical fan curves provided are generally the results of wind tunnel test under ideal conditions with close tip clearances of 2mm - 4mm without inlet / outlet restrictions.

Depending on tip clearance, cowl design, fan / core location, depth and height and width, radiator inlet restrictions, outlet restrictions, and fan performance may have to be derated by a minimum of 20% and quite frequently up to 50%.

Therefore, the air flow stated on the design data sheet must be proven with the radiator installed at the first stage of a cooling test and trial.

The Anemometer is probably the most practical test piece available.



11. From the radiator designers view point, his responsibility should end by stating the air flow and PD required to meet the duty specified.

This of course is somewhat impractical.

To ensure that the proper options available are taken, the radiator designer should be provided with a fan curve and fan drive ratio to ensure that the air flow required is within reasonable limits of the fan.

However, there are numerous aspects of the installation that the radiator designer cannot foresee - most of which are not submitted in the initial enquiry - all of which relate to the installed fan efficiency.

The selection of a fan for the application is often limited to both technical and commercial (availability) reasons.

12. Steel blade fans are commonly used on diesel engines for automotive and general use, however with increasing emphasis on noise levels and silencing, the aerofoil section fan is taking precedence.

Aerofoil fans will operate at higher duty (PD) levels and on a straight comparison basis, will deliver more CFM / parasitic HP than that of the steel blade units.

They are generally more efficient provided that the installation is compatible.

We will cover this area later in Mechanical Design.

Therefore, in simple terms the radiator is a function of mass flows, temperature differentials, and cooling surface area. - But practical terms, is more complex.

13. The important aspects are the input data provided to the radiator designer, and we now request your involvement by looking at the Design Data Sheet.

Apart from commercials, such as names, dates, file and reference numbers

(i) Engine / model required for record purposes.

(ii) Gross rating and RPM - a number of engines are basically the same with different fuel settings, therefore again for record, it is helpful to know the precise model we are dealing with.

(iii) Design BHP (KWB) and RPM - numerous special applications use only 70 - 90% of available gross HP, and therefore, the heat rejection would be less.

For generating sets, the engine BHP at 100% load should be stated, not the KVA or KWE rating - we are not to know the alternator efficiency.

For manual trucks - two cooling calculations are necessary :-

- (a) Maximum torque or minimum speed range.
- (b) Maximum governed speed.

Depending on fan / radiator / engine on most units, the maximum torque condition is 'hardest' to cool, as the amount of air available is less at lower speed than at maximum governed speed - also with flat torque engines the heat rejected is almost constant.

Automatic (torque converter) units - trucks or mobile equipment.

The design point is at the converter engine match point, which relates engine input

and transmission output and duty. Depending on duty, this may be from 60 - 90% efficiency.

This match point is variable to each engine - torque converter - transmission application and generally falls within the engine maximum torque RPM and maximum governed RPM.

Note Each unit must be matched, there is no opportunity for guess work.

Portable Compressors - or diesel driven hydraulic systems that use cooling air where more than one cooling circuit is involved, requires special consideration.

Depending on the heat rejections, maximum temperatures of both liquids and the air flow direction, a radiator / oil cooler combination is quite common.

However, the standard radiator, fan combination can very rarely be used due to the air rise and PD over both coolers.

## 14. Terminology

Ambient Air - as far as the radiator design is concerned, is the air temperature at point of entry to the core.

It is not the outside air temperature or that temperature that your customer might specify for the piece of equipment.

This requires some explanation :-

1. With the suction fans (air flow), if the air flow is induced directly from outside, then the ambient is most likely the same.

2. With blower fans (air flow) in remote radiators outside power stations, it is the same.

3. If the radiator is mounted inside the power station, or a closed canopy or the like, then a temperature rise of air must be encountered from outside ambient, due to radiated heat from the equipment.

Diesel engine exhaust radiation, alternator heat, transmission heat, compressor heat etc., incur heat to flow from them to the surrounding air, as the air for the radiator is induced over these areas, then an air temperature rise must result.

We have recorded up to 36°F (18°C) rise.

Therefore, this must be specified as a separate rise over ambient.

Heat Rejection:

Varies with different engines.

From 16-60 BTU/BHP/MIN for water and similarly for oil and intercoolers - do not assume all engines are the same.

Transmission and hydraulic systems are related to that machines efficiency.

Flows:

Particularly with oil and its permissible pressure drop at the rated design temperature are important. With differing options for fan ratios, most water pumps offer differing characteristics, these flows must be stated accordingly.

Max. Temp.

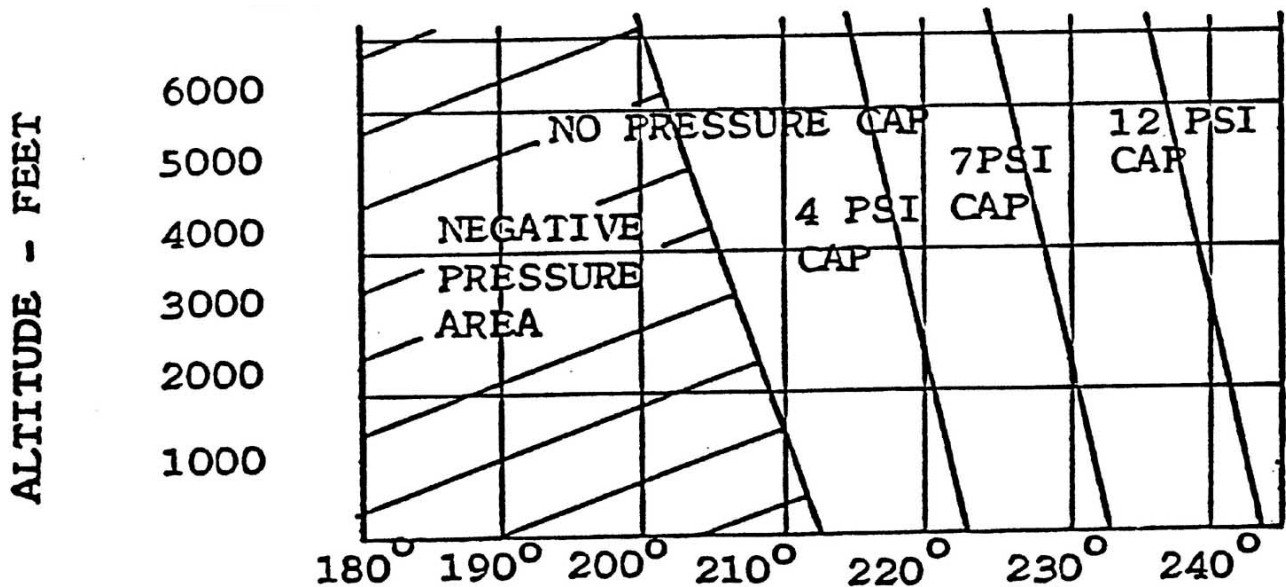
Does not mean boiling.

It means the maximum allowable temperature of the liquid under design conditions.

JW temperatures vary from 140°F - 230°F  
Transmission Oil                      250°F - 300°F

Hydraulic Oils from 130°F - 190°F  
 Charge water is generally 15°F  
 approach to ambient.  
 (More recently 10°F (5°C)).

15. The Chart shows the temperature at which water boils at varying pressures and altitudes.



Note: Air to boil? The air temperature at which boiling occurs at sea level without a pressure cap is 212°F. Boiling has nothing to do with our calculations.

### Fan Diameter Ratio:

Performance curves and options are a vital piece of design information, and regrettably are the hardest thing to have made available to us.

It can be seen however, that the fan plays an important part.

Do not expect a standard fan - radiator to operate in noise controlled generating sets, compressors, etc., in canopies. Air

pressure drops of up to 02" w.g. inlet and +1" w.g. on outlet, have been encountered, add the radiator core of say 1" w.g. and the fan is expected to work in 4" w.g. - this is not possible, the fan will stall.

Therefore, any information of a general nature must be forwarded with the enquiry.



## Output Data:

In our quotes at the bottom of the sheet, you will see the core or radiator selected together with the face velocity, that must be achieved if the unit is to operate to design conditions.

It is important to recognise the air flow stated must be achieved.

## 18. Proving and Testing

Air flow - with remote radiators where we control the fan - fan tip clearance, etc., we prove our air flow. On radiators where the fan is of customers supply and installation, it is clearly the customers responsibility to ensure that the air flow stated is being met.

Most of you would appreciate we have seen cases that are almost beyond belief.

They vary but here are some:-

(a) Double walled canopies, air PD 2.5" w.g. and cores.

(b) Fan tip clearances of up to 4".

(c) No cowls - or poor fitting.

(d) Blower fans located 1.5" from core.

(e) Hot air recirculation from sides and bottom and top of radiators (both blower and suction fans).

(f) Hydraulic oil coolers mounted in front of radiators - not included in design information.

(g) Part of core covered by vehicle cabs or plumbing.

(i) On V engines in COE vehicles no exit air capacity - its all engine and no space for air to flow.

(j) 4" or 6" channel immediately in front of radiator core.

(k) Incorrect fan drive ratios.

(l) Incorrect fans.  
and many more.

Having established that the air flow is being met, a heat load test should be conducted with the thermostat jammed open.

Details of test procedure are covered in later T.I.B.'s.

**For further information, contact Air Radiators.**

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