Optimum Heat Recovery from Diesel Engines for Heat Supply of Fresh Water Generators on Cruise Ships.

Most of the actually cruising large passenger ships are equipped with diesel-electric drives. The future will show, if the new drive concept including gas turbines ore even the combination of gas turbine and steam turbine will prove to be more economic. In this presentation the author concentrates on the conventional diesel-electrically driven cruise vessels.

Diagram 1 shows the typical simplified motor water flow diagram of a Diesel engine. The Motor Water is pumped with a directly or electrically driven pump through the engine, due to the Diesel-electric concept mostly with a constant flow rate. The engine outlet temperature is controlled by an automatic recirculation of heated motor water back to the engine inlet, typical controller setpoint is 90 °C. The engine inlet temperature $T_{ENGINE\text{-}IN}$ can be calculated as follows:

$$T_{ENGINE\text{-}IN} = T_{ENGINE\text{-}OUT} - Q_{HT} / (M_{COOL} * CP)$$

with:

	CP	= specific heat of the motor water as (approx. 4.18 kJ/kg K).
	M_{COOL}	= motor water flow through engine
	$T_{ENGINE\text{-}OUT}$	= motor water temperature outlet engine (control)
	Q_{HT}	= exhaust heat in HT motor water

The residual (not remixed) motor water is flowing to the engine cooler, which – in this example – is operated with seawater directly. The engine cooler is designed for certain worst case conditions, for example 105% engine load at tropical conditions. In this critical operating point the minimum achievable cooler outlet temperature must be equal to the engine inlet temperature necessary for cooling. For all other operating points the cooler is over-dimensioned and will cool the motor water down to a temperature corresponding to the heat transfer capacity of the cooler. Diagram 2 shows typical exhaust heat characteristics of a Diesel engine.

In case of no further control loop is installed, the motor water return temperature behind the cooler ($T_{COOLER\text{-}RETURN}$) will drop – at declining load of the engine – closer and closer to the temperature of the cooling medium. This results in a correspondingly low residual motor water flow (M_{COOLER}) being needed for cooling, which can be calculated with the following formula:

$$M_{COOLER} = Q_{HT} / (CP \times (T_{ENGINE\text{-}OUT} - T_{COOLER\text{-}RETURN}))$$

In case of connecting a freshwater generator (evaporator) to the motor water cycle which returns the motorwater with the temperature $T_{EVAP\text{-}RETURN}$, the portion of recovered heat from the existing exhaust engine heat can be calculated easily as follows:

$$Q_{RECOVERY} / Q_{EXISTING} = (T_{ENGINE\text{-}OUT} - T_{EVAP\text{-}RETURN}) / (T_{ENGINE\text{-}OUT} - T_{COOLER\text{-}RETURN})$$

Diagram 3 and 4 show the behaviour of motor water flow and temperature within the load range of a Diesel engine, in case of an uncontrolled cooler (with no bypass). Diagram 5 shows the corresponding achievable heat recovery, in case of the evaporator returns the motor water to the engine cooler at a temperature of 73 °C.

If we take read for example the curves at an engine load of 85%, we have the following situation:

$T_{ENGINE-OUT}$: 90 °C , $T_{EVAP-RETURN}$: 73 °C , $T_{COOLER-RETURN}$: 60 °C

The percentage of heat recovery – according to the equation given above - is:

$Q_{RECOVERY}$ / $Q_{EXISTING}$ = ((90 – 73) / (90 – 60)) * 100% = <u>56.7 %</u>

At 85% engine load (at tropical conditions) in this example only 56.7 % of existing HT motor water exhaust heat are recovered, while 43.3 % get lost in the uncontrolled engine cooler.

The calculation example as well as the curves show clearly that – for optimisation of the heat recovery – the cooler return temperature to the engine should be raised to a level being as close as possible to the engine inlet temperature. In case of – even at full load of the engine – all available heat shall be used for the evaporator - the only way to maintain 100% heat recovery is to keep the engine inlet temperature required for cooling at the same temperature at which the motor water is returned from the evaporator (typically 72 – 73 °C). In many cases the geometry of the engine's cooling channels does not allow the correspondingly high motor water flow rates, or the pumping capacity – for cost reasons – has to kept below a certain economic maximum. In most cases – on cruise ships – the recoverable HT-motor water heat – at full engine load - exceeds the required heat for the fresh water generator so that a certain heat rejection in the cooler can be tolerated. But it remains important to improve the heat recovery at part load operation of the engines which is more often the case than full speed cruising.

A practicable way to raise the motor water return temperature behind cooler is to install a second controller with a motor water bypass in parallel to the engine cooler in order to reduce the effectiveness of the cooler at part load operation (please refer to diagram 6). An important question is now how to adjust the setpoint of this controller behind. On one hand the setpoint of this controller should be as close as possible to the engine inlet temperature required for cooling, because in this case the recirculation flow at the engine is low and net cooling water flow to the cooler utilisable for heat recovery is correspondingly high. On the other hand there should always remain a certain safety margin below the engine inlet temperature in order to avoid superheating of engine at sudden load changes.

The most simple control method way is to adjust the setpoint at a fixed value covering the worst case operating condition (for example 105% load operation at tropical conditions). This method may lead to an incomplete utilisation of the engine's exhaust in the connected evaporator, especially in case of engines with a low motor water pump capacity.

A more "intelligent" control method is to raise the cooler return temperature with declining engine load. This can be done by establishing in the computerised control system a curve with a certain dependency of the cooler temperature setpoint of the engine load. Diagram 7 shows different curves for the cooler temperature setpoint.

Diagram 8 shows the effect of these different setpoint curves on the net motor water flow to the cooler, from which the connected evaporator can divert the required thermal energy. Diagram 9 shows the effect of the cooler temperature control on the heat utilisation (at return temperature from evaporator = 73 °C). The main result is that the constant value control method leads to a still very ineffective utilisation of the engine's exhaust heat. A load-dependent adjustment of the cooler temperature setpoint is strongly recommendable. The best results gives the curve 3, where the setpoint is adjusted according to a linear function between 100% and 50% load and remains constant below 50% engine load, with a safety margin of 2K (in this example) from the theoretically required engine inlet temperature. The diagrams 7 – 9 are valid for tropical operating conditions. The effectiveness of the control is still more significant in case of ISO-conditions.

Summary:

Most of the existing large cruise vessels have diesel-electric drives and use evaporators as fresh water generators which consume a lot of thermal energy. For better heat utilisation of the HT-motor water heat from Diesel engines the motor water flow through the engines should be selected as high as possible and tolerated by the engine supplier. Engines with higher motor water throughputs are preferable with respect to completeness of heat utilisation.

In case of motor water temperatures at engine inlet below 72°C (which is a typical motor water return temperature from the evaporator) it is useful to adjust the setpoint of the cooler temperature controller in dependence of the engine load (rising setpoint with declining engine load). The efficiency of heat recovery is significantly improved, and a lot of steam can be saved (1000 kW = approx. 1590 kg/h of steam).

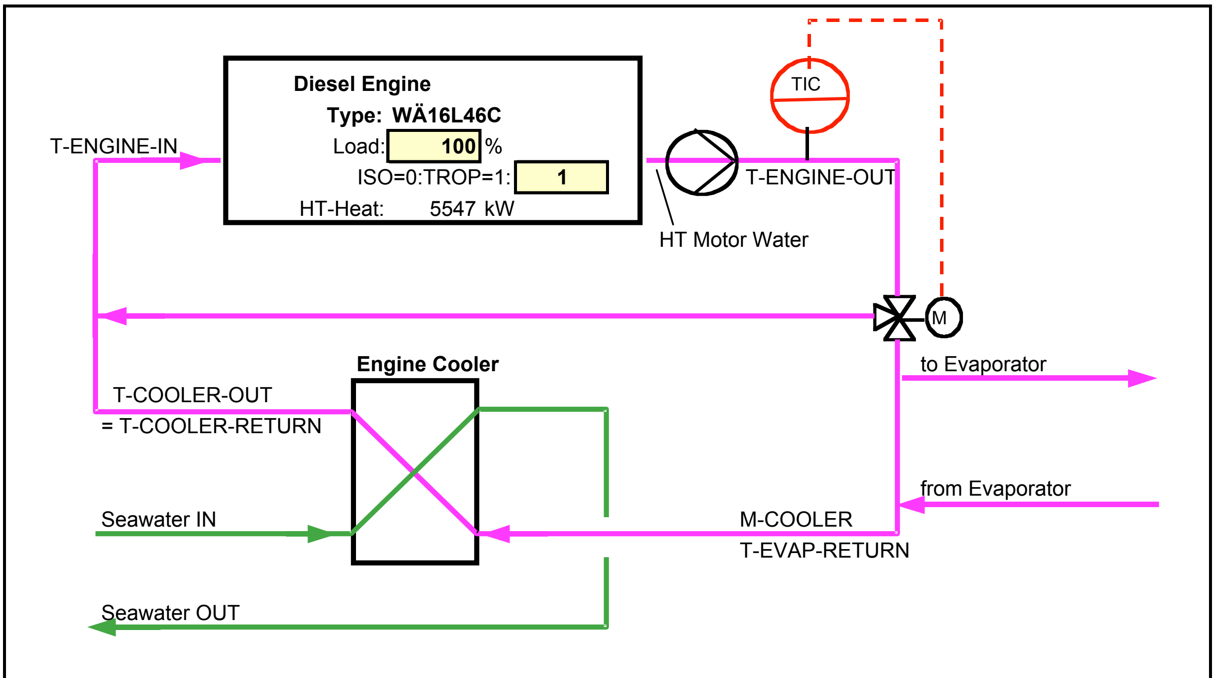
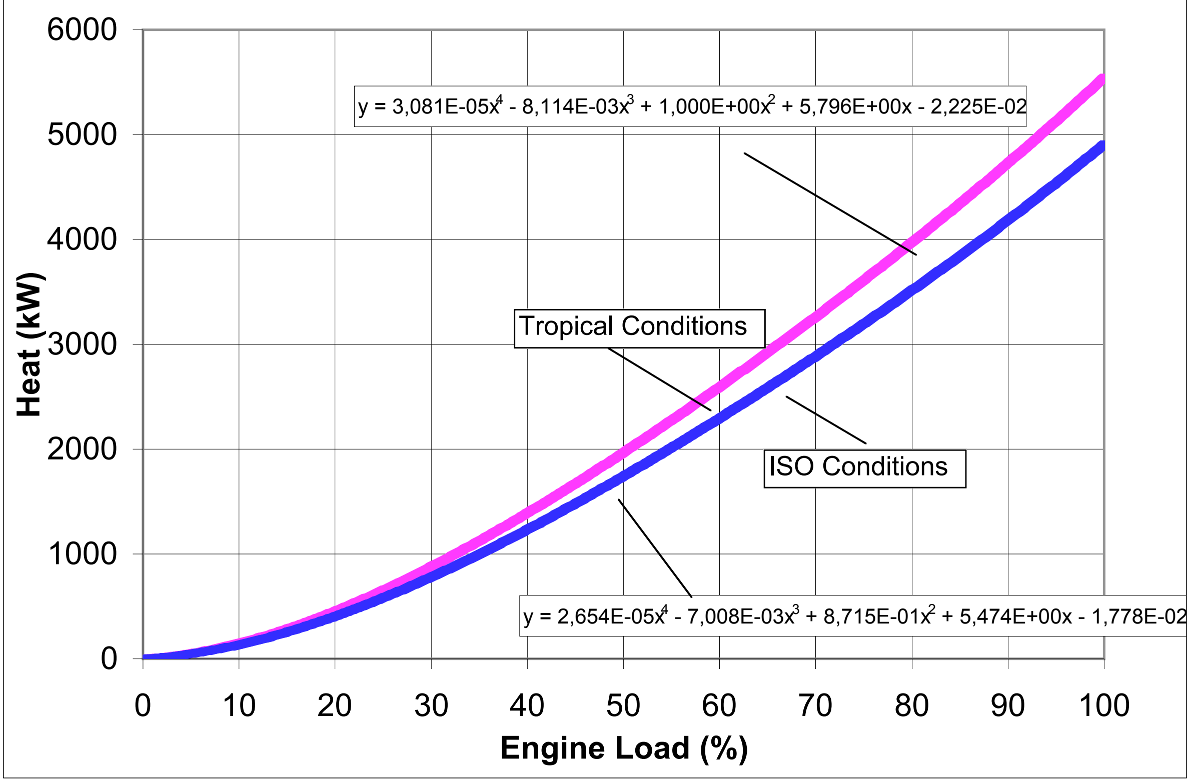


Diagram 1: Diesel Engine with Cooler and Temperature Control

Diagram 2: HT-Motor Water Heat of Wärtsilä 16L46C Engine



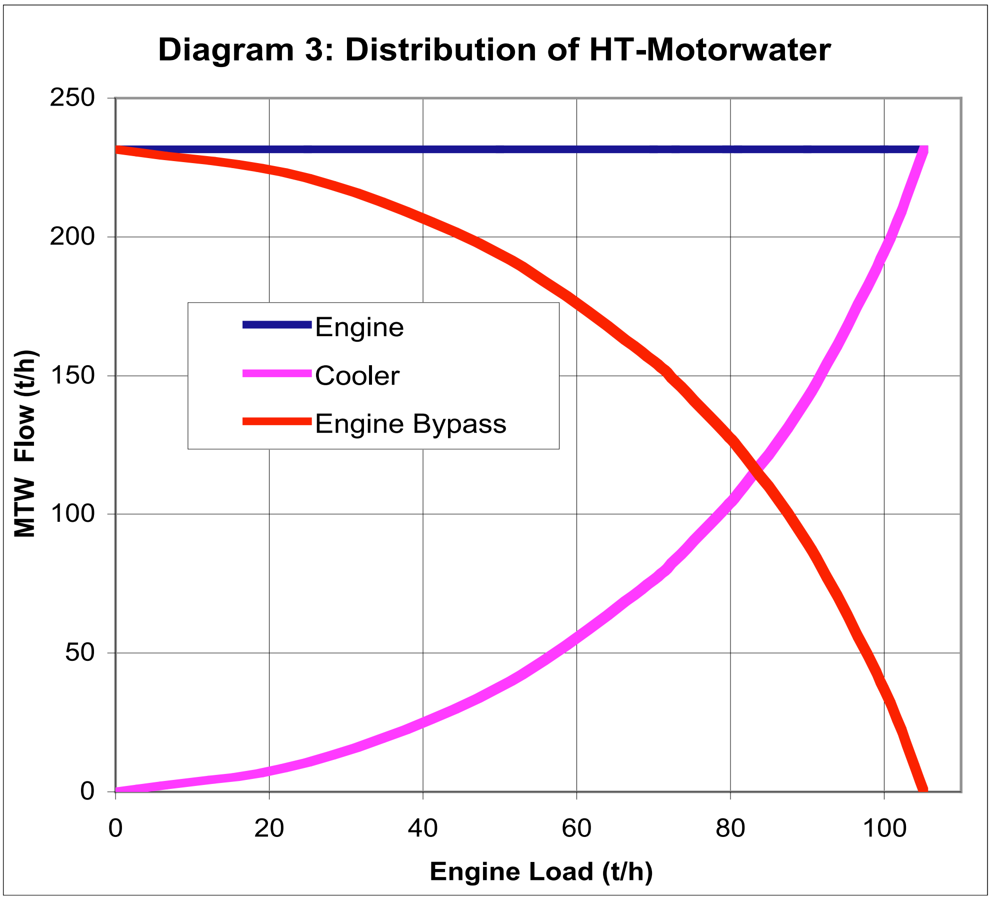


Diagram 3: Distribution of HT-Motorwater
MTW Flow (t/h)
250
200
150
100
50
0
0
20
40
60
80
100
Engine Load (t/h)
Engine
Cooler
Engine Bypass

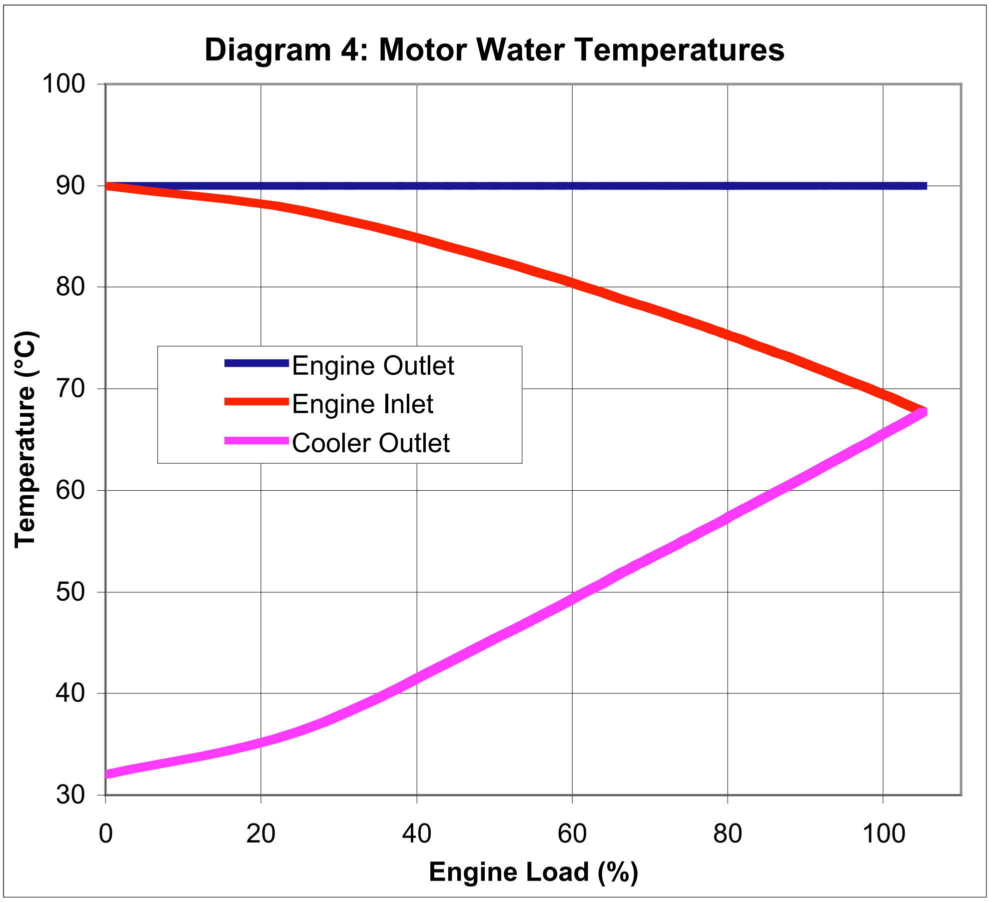
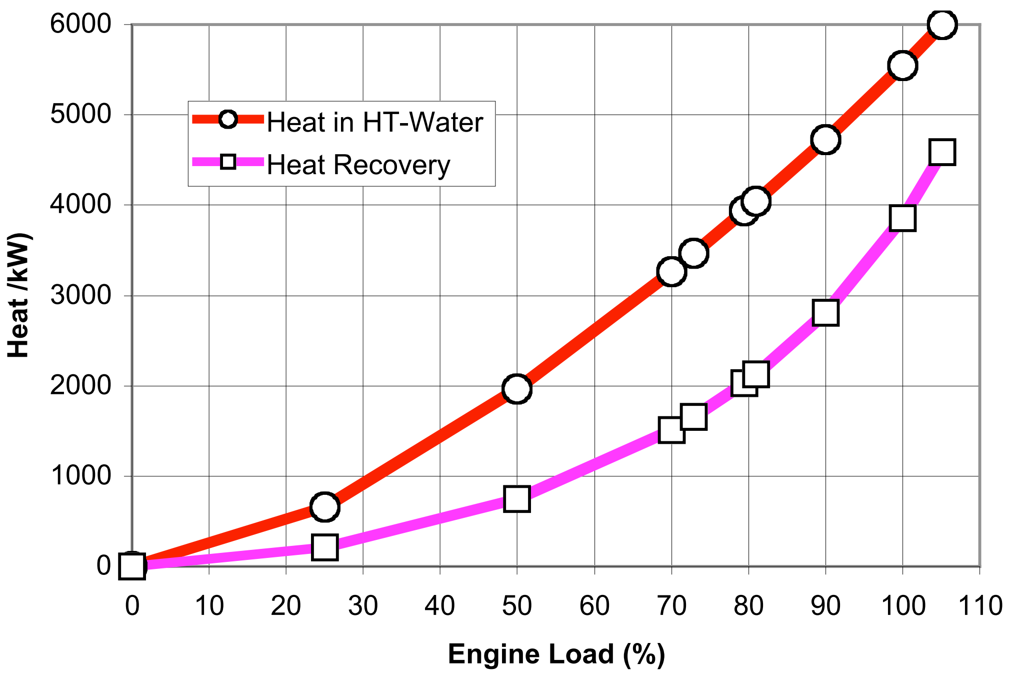


Diagram 4: Motor Water Temperatures
Temperature (°C)
100
90
80
70
60
50
40
30
0
20
40
60
80
100
Engine Load (%)
Engine Outlet
Engine Inlet
Cooler Outlet

Diagram 5: Heat Recovery from Motor Water



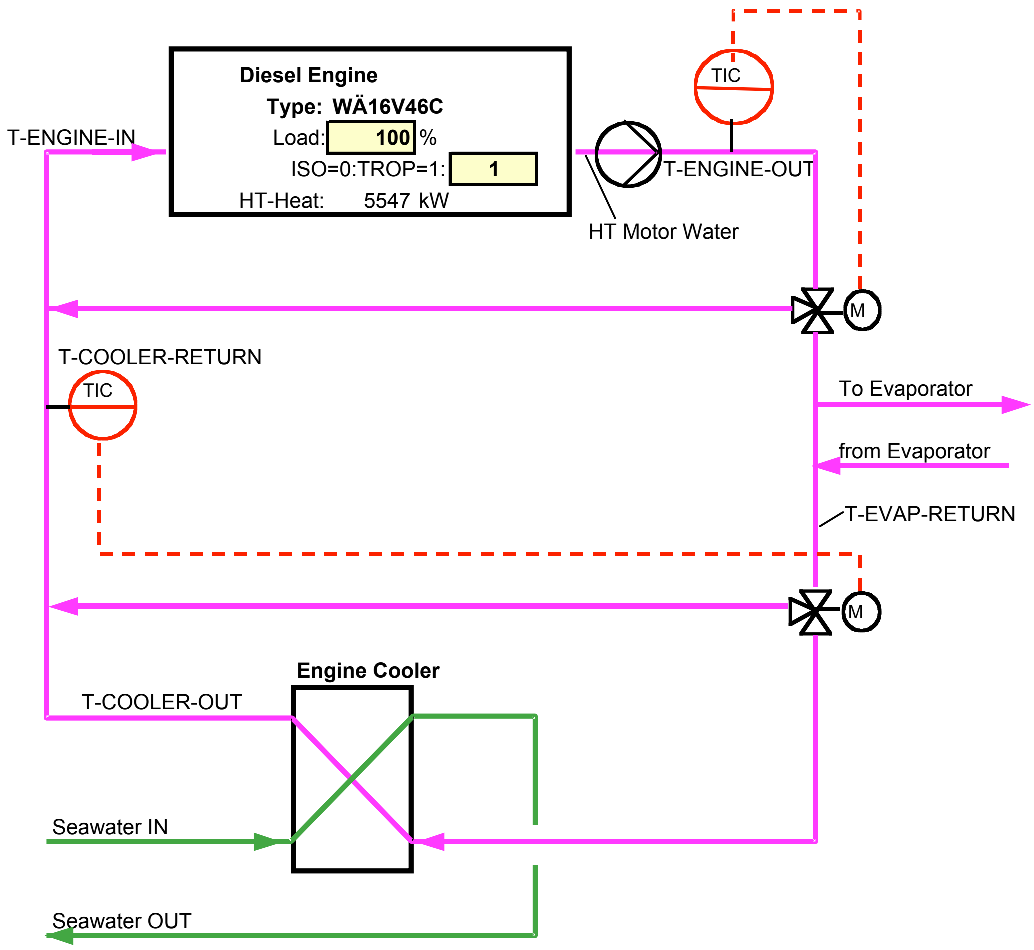


Diagram 6: Diesel Engine with Cooler and Temperature Control

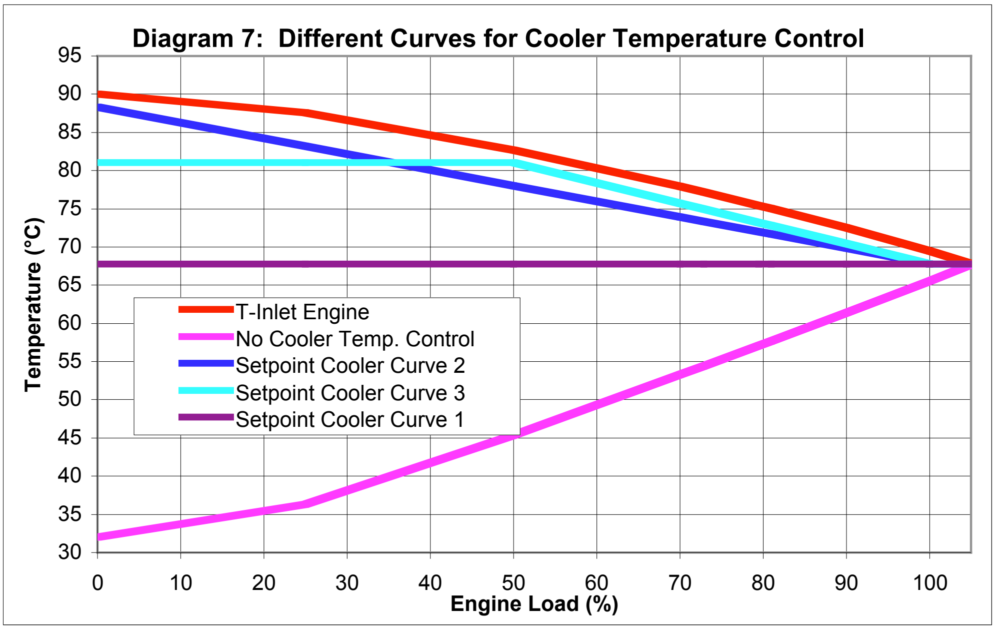


Diagram 7: Different Curves for Cooler Temperature Control
Temperature (°C)
95
90
85
80
75
70
65
60
55
50
45
40
35
30
T-Inlet Engine
No Cooler Temp. Control
Setpoint Cooler Curve 2
Setpoint Cooler Curve 3
Setpoint Cooler Curve 1
0
10
20
30
40
50
60
70
80
90
100
Engine Load (%)

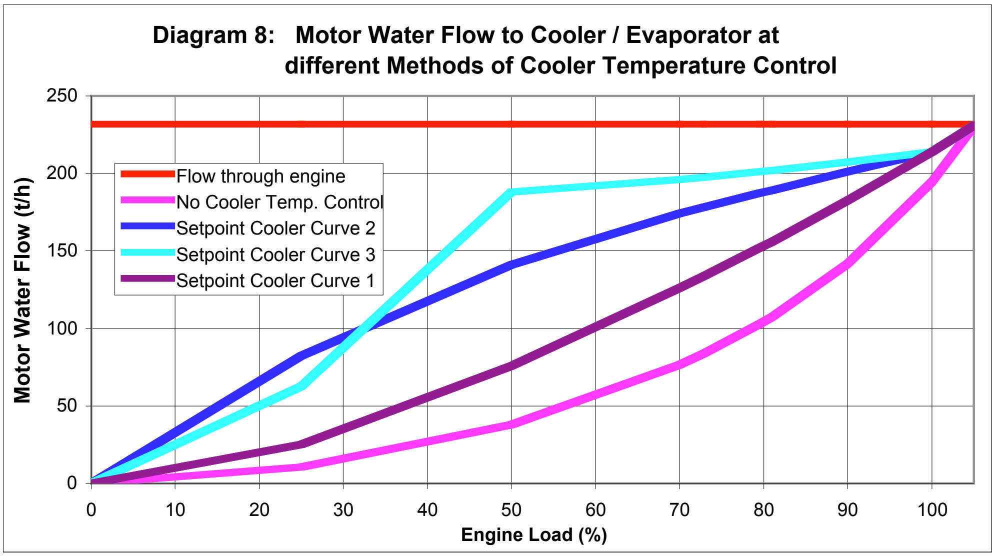


Diagram 8: Motor Water Flow to Cooler / Evaporator at different Methods of Cooler Temperature Control
Motor Water Flow (t/h)
250
200
150
100
50
0
Flow through engine
No Cooler Temp. Control
Setpoint Cooler Curve 2
Setpoint Cooler Curve 3
Setpoint Cooler Curve 1
0
10
20
30
40
50
60
70
80
90
100
Engine Load (%)

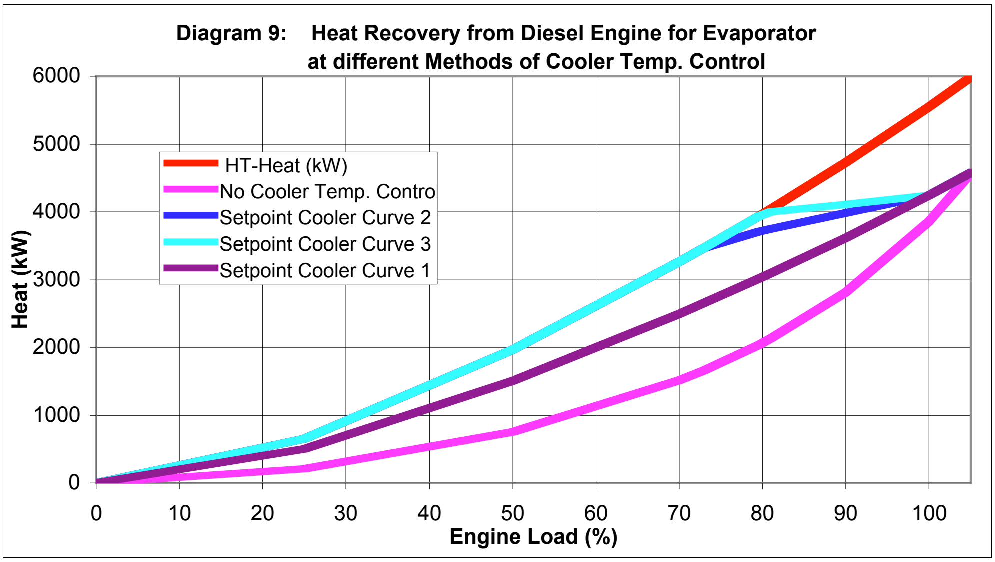


Diagram 9: Heat Recovery from Diesel Engine for Evaporator at different Methods of Cooler Temp. Control
Heat (kW)
6000
5000
4000
3000
2000
1000
0
HT-Heat (kW)
No Cooler Temp. Contro
Setpoint Cooler Curve 2
Setpoint Cooler Curve 3
Setpoint Cooler Curve 1
0
10
20
30
40
50
60
70
80
90
100
Engine Load (%)