Electric turbocharging – A path to increased lean burn gas genset efficiency together with diesel like transients

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Abstract: Improvements in high speed electrical machine technology and power electronics have enabled the development of many new electric turbocharging concepts for internal combustion engines, including electric turbines, electric compressors, and electric turbochargers.

This paper collates Bowman Power Group's experience in designing and applying these technologies within the power generation and heavy-duty truck markets and compares and contrasts each technology specifically for a high-speed natural gas genset. Results, based on a mix of measurement and simulation, show that through careful sizing of the turbocharger's turbine it is possible to use electric turbocharger concepts to increase full load electric efficiency of a modern gas genset by up to 2.0% points and provide transient load performance akin to that of modern emergency standby diesel gensets.

Key Words: Electrification; electric turbine; electric compressor; electric turbocharger; heat recovery; transients; emissions

1.0 Introduction

As regulations and market demands continue to change, the requirements on high-speed internal combustion engines continue to push OEM's to design engines capable of:

- higher power density (brake mean effective pressure BMEP)
- lower fuel consumption
- lower emissions
- greater operational flexibility in terms of
  - ambient conditions
  - fuel quality
  - transients

In order to achieve efficiency and BMEP competitiveness, it is common practice for medium and high speed gas engine manufacturers to use rapid lean burn combustion concepts which require ever increasing levels of Miller cycle to keep emissions, in-cylinder conditions and temperatures within safe working limits[7]. Increases in turbocharger (TC) pressure ratios and efficiencies have enabled this, providing the high boost pressures required to tread the fine line between combustion knock and misfire at high Brake Mean Effective Pressure (BMEP) (Figure 1) without significantly impacting the backpressure or scavenging of the engine.

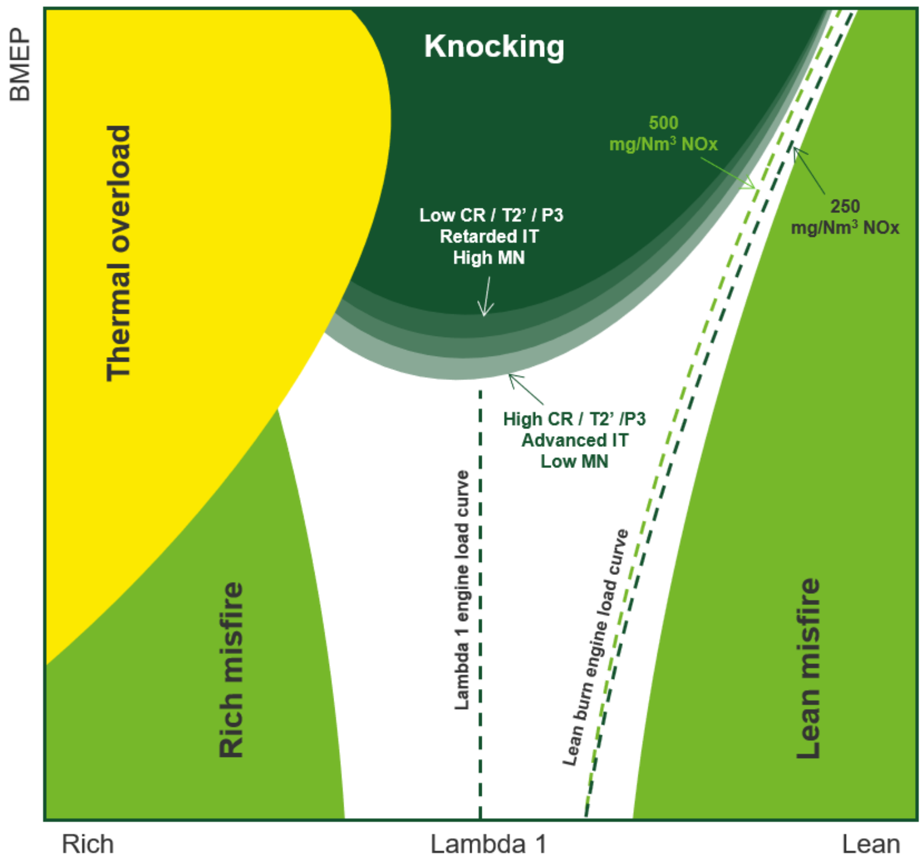

Figure 1: Combustion boundaries for high BMEP gas engines

The high pressure / efficiency turbocharging requirement for lean burn gas engines results in TC designs and layouts which traditionally have high inertia, and low efficiencies away from the full load design point versus an equivalently sized diesel engine (Figure 2). These factors, together with combustion constraints (knock and misfire) that limit the ability to aggressively enrich, result in extremely poor transient load acceptance capability when compared to diesel engines.

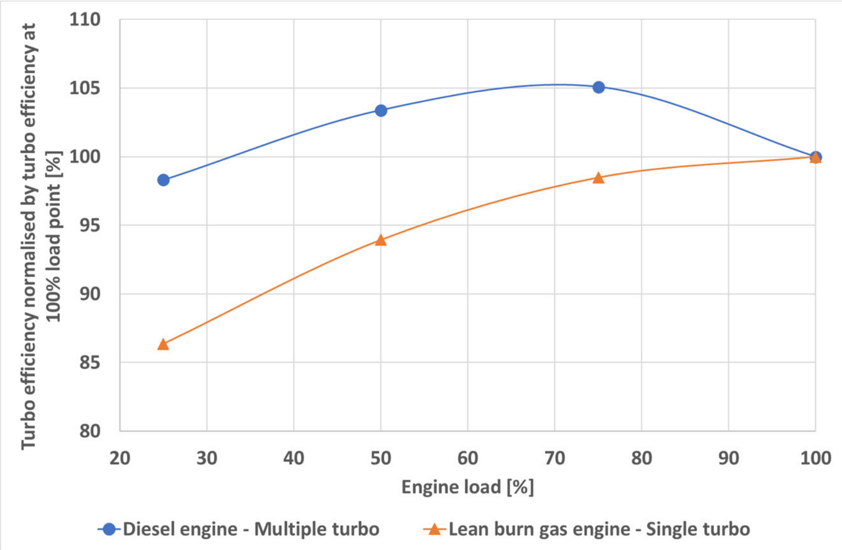

Figure 2: Normalised turbocharger (TC) efficiency versus engine load

These challenges primarily limit the application of high BMEP, high efficiency lean burn gas engines to base load power generation, which have aggressive BMEP and fuel consumption requirements but modest transient ones. Opportunities for lean burn gas engines to diverge into other markets (even within power generation) are relatively limited without considering radical changes in technology adoption.

Bowman Power Group (BPG) have for over 16 years specialised in designing, developing, and manufacturing turbomachinery coupled to high speed electrical machines (HSEMs) and the power electronics (PE) to drive and control them.

BPG's HSEM machine portfolio fits within the 13 – 270kWe power and 20 to 120krpm speed ranges (Figure 3).

As can be inferred from this range of power and speed capability, BPS's HSEM's have been applied to turbomachinery covering a wide range of engine applications (HD trucks, tractors, Powergen, Rail), sizes (150 – 18000kW) and fuels (diesel, biodiesel, natural gas, biogas, etc) providing valuable learnings and application experience when optimizing these technologies to meet the plethora of OEM and market requirements [8, 9, 11, 12].

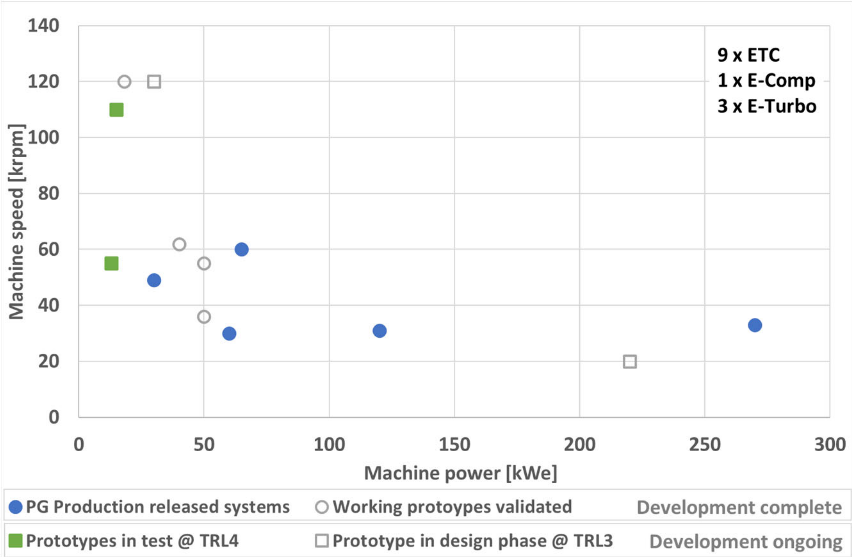

Figure 3: Bowman Power Group HSEM Turbomachinery portfolio

Three HSEM turbomachinery topologies have been developed and manufactured:

- Electric turbo compounding (ETC) where a turbine coupled directly to an electric generator is placed downstream of the engine's TC
- Electric compressor (E-Comp) where a compressor coupled directly to an electric motor is placed within the intake system
- Electric turbocharger (E-Turbo) where an electric motor / generator is placed within the main engine TC body

A high-level schematic of these HSEM technologies can be seen in Figure 4. As is well documented in literature, each has specific strengths in allowing the turbocharging system to be optimised with the aim to improve fuel consumption by means of heat recovery (generating with the HSEM and PE) [1, 3, 6, 14] and / or improve load response and reduce emissions (motoring the PE & HSEM) by rapidly increasing boost pressure [6, 9, 10, 13].

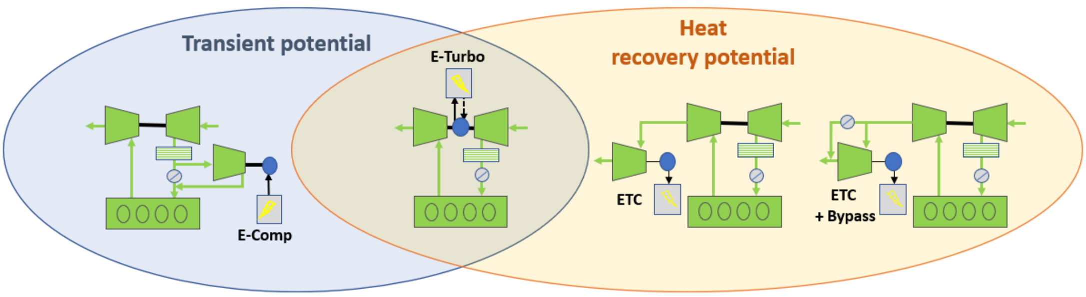

Figure 4: HSEM layouts investigated for heat recovery and transient improvements

This paper documents the steady state efficiency benefits achievable with ETC and E-Turbo and the transient benefits achievable with E-Turbo and E-Comp when optimised and applied to a modern high seed, high BMEP, lean burn natural gas genset.

2.0 Efficiency improvements through heat recovery

2.1 Heat recovery /pumping trade off

The theory of the heat recovery / pumping loss trade-off for ETC is well documented [2] and has been expanded upon further looking specifically at the effects on spark ignited engines.

Figure 5a shows a representation of the energy available in the exhaust gases of a turbocharged gas engine from Exhaust Valve Opening (EVO, which is approximately equal to bottom dead centre (BDC)) through to the exhaust gas leaving the stack of the engine at atmospheric pressure. The energy available can be split into three main components:

1. the blow down energy from the high cylinder pressure pulse expelled at the time of EVO
2. the energy available through expansion from the exhaust manifold (P3) to atmosphere
3. the energy generated by the piston during the exhaust stroke (moving from BDC to Top Dead Centre (TDC) which is approximately equal to the time of Exhaust Valve Closing (EVC))

As high-speed gas engine exhaust manifolds are designed as constant pressure systems, then the energy recovery potential from the blow down pulse is limited. Therefore, the exhaust energy available for the turbocharging system can be simplified to the sum of the exhaust energy available at EVO (expanding from P3 to atmospheric pressure), plus that generated by the piston (moving from BDC to TDC). In this case the TC turbine is sized such that P3a is the exhaust pressure required to generate the compressor power and boost pressure necessary to achieve 100% load at ISO conditions without throttling the engine.

Figure 5b shows a real-world application where the TC is matched more aggressively so that governing reserve (additional boost pressure) is available to account for potential high altitude and high ambient running conditions, as well as engine ageing and fouling effects through the life of the engine. In this case the additional power requirement of the TC drives a higher exhaust pressure, P3b and thus a loss in engine efficiency due to the additional pumping work done by the piston versus 5a. This should be regarded as the baseline configuration.

Figure 5c shows the situation when the TC turbine nozzle area is further reduced to increase the expansion ratio across the turbocharging system and therefore extract more power from the exhaust gases. The increase in exhaust pressure to P3c results in additional pumping work for the piston, negatively impacting the engine efficiency. However, by applying ETC downstream of the engine's TC, the additional exhaust energy available can be recovered and converted to electrical power. When factored by the ETC turbine isentropic, shaft, generator and PE efficiencies, the electrical power generated is in the order of 2 to 2.5 times the pumping power lost for a typical gas engine. This gives a net electrical efficiency benefit when looking on a system level.

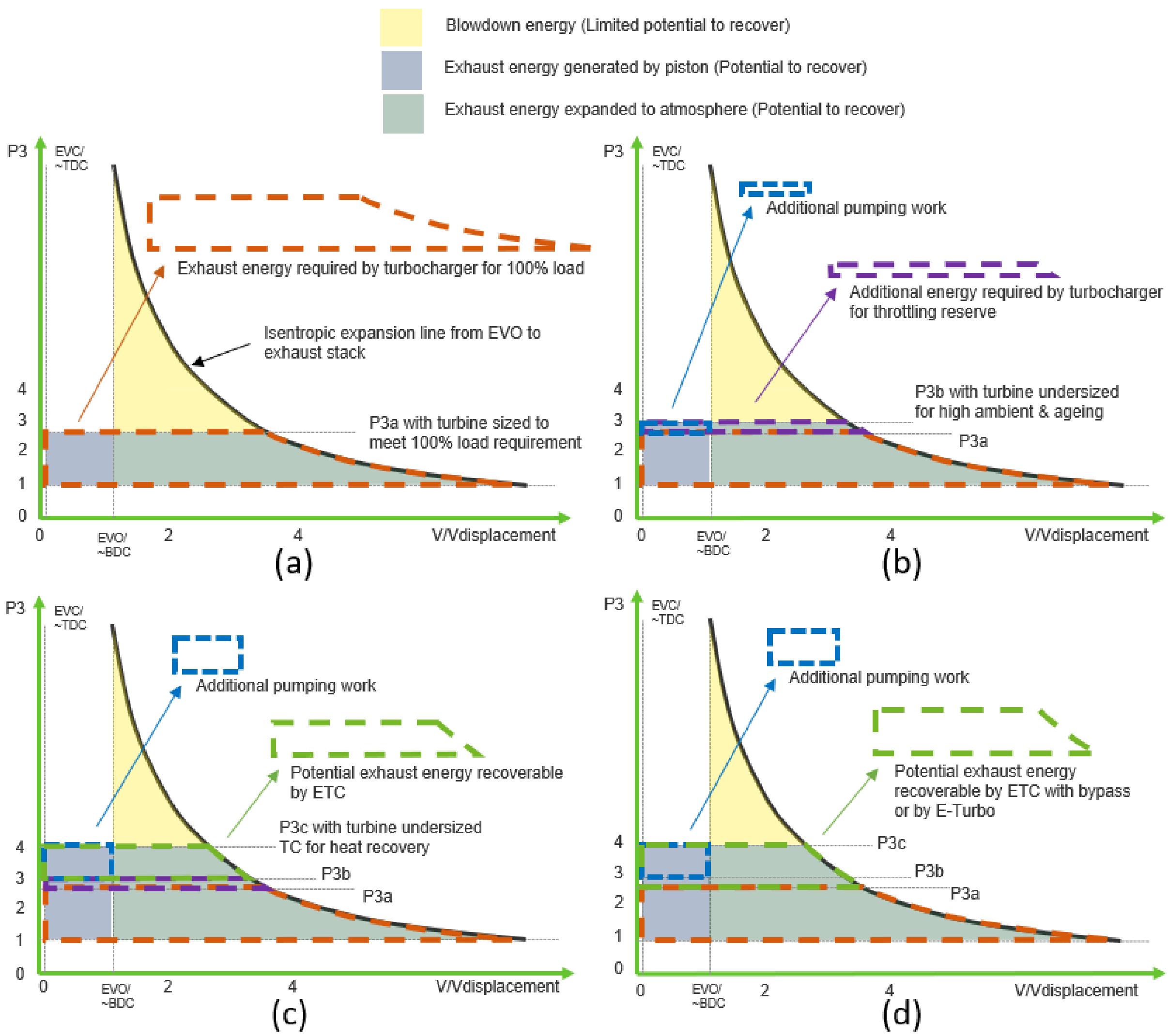

Figure 5: Exhaust energy required by turbocharging system of high-speed gas engine. (Images expanded upon from ABB publication [2])
(a) TC matched exactly at ISO conditions
(b) TC matched in real world application, with additional governing/throttle reserve for high ambient running with aged engine (Baseline engine configuration)
(c) TC system matched for heat recovery with governing reserves for high ambient running with aged engine
(d) TC system matched for maximum heat recovery at ISO conditions

If the boost pressure of the engine is controllable by adjusting the expansion ratio on the ETC turbine, then the power recovered from the TC system can be further enhanced. This can be achieved by reducing the ETC turbine area to de-throttle the engine under ISO conditions (Figure 5d). This is made possible by integrating a bypass valve (Figure 4) to flow exhaust gas around the ETC / decrease the backpressure on the TC as ambient temperatures increase. This ensures the system efficiency can be maximised at ISO conditions while ensuring 100% system load can be maintained at temperatures above 25°C, albeit with reduced net electrical efficiency benefit.

The same effect can be achieved by using an E-Turbo to recover the excess exhaust energy rather than using ETC. In this case the load being generated by the E-Turbo HSEM can be controlled to target the minimum governing reserve needed to achieve 100% load at ISO conditions. 100% load can then be achieved at higher ambient temperatures by reducing the power generated by the E-Turbo.

With the same aerodynamic TC match and piston pumping losses, an E-Turbo will result in a marginally lower electrical efficiency benefit versus ETC due to there being only one expansion process from P3c to atmospheric and thus less electrical power recovered.

2.2 Scavenging and combustion considerations

The high efficiency TC's used in high BMEP high-speed gas engines result in positive scavenging pressure (Intake manifold pressure (P2') – P3)) at full load. Depending on the baseline engine layout, and how aggressively the TC nozzle area is downsized to enable heat recovery (as described above), the scavenging pressure will reduce towards zero and in some cases may even become negative. This has three important influences worthy of discussion:

2.2.1 Exhaust gas residuals effect on knock

Decreasing the scavenging pressure has a negative effect on spark ignited combustion through increased exhaust gas residual fraction. Exhaust gas molecules act as free radicals, promoting faster combustion reactions and speeds, as well as increasing the bulk charge temperature. Both lead to increases in the pressure and temperature of the fuel / gas mixture in front of the flame front on a crank angle basis, versus the baseline engine, which increases the propensity of end-gas knock.

When the TC turbine area is reduced, other actions must be taken with the engine settings to maintain the same margin to end-gas knock with the same fuel composition as the baseline engine. This can be achieved through any combination of derating the engine load (this can be offset against the additional power produced by the ETC or E-Turbo) and / or reducing piston compression ratio and / or retarding spark timing. All will result in a loss in engine efficiency and must be considered and traded when downsizing the TC turbine area to increase the heat recovery potential of the TC system.

2.2.2 Fuel short circuiting / methane slip

High speed gas engine fuel is normally mixed upstream of the TC compressor, leading to the potential for significant fuel slip during valve overlap. Even if the valve overlap area is small by design, significant fuel short circuiting can occur as valves and seats wear driving Intake Valve Opening (IVO) and EVC further apart between service intervals / lash adjustments. Typically, when applying ETC, the TC turbine area needs to be reduced such that the scavenging pressure decreases by 0.7 to 1.0bar. This is necessary so that ETC turbine expansion ratio achieved is high enough to achieve good isentropic efficiency. Decreasing the scavenging pressure below 0.2bar eliminates [4] or at least significantly reduces, methane slip occurring during valve overlap as illustrated in Figure 6.

Reducing the scavenging pressure additionally lowers the proportion of the unburnt hydrocarbons (UHC) from the cylinder and piston crevices entering the exhaust port. Although the influence of the scavenging pressure on reducing the crevice UHC entering the exhaust is believed to be small in relative terms, it is still worth noting as the UHC trapped and shielded from combustion within the crevices can be significant in absolute terms (between 1 and 3% of the total fuel consumption depending on the piston design philosophy), the majority of which is expelled as the piston reaches top dead centre (TDC), during valve overlap.

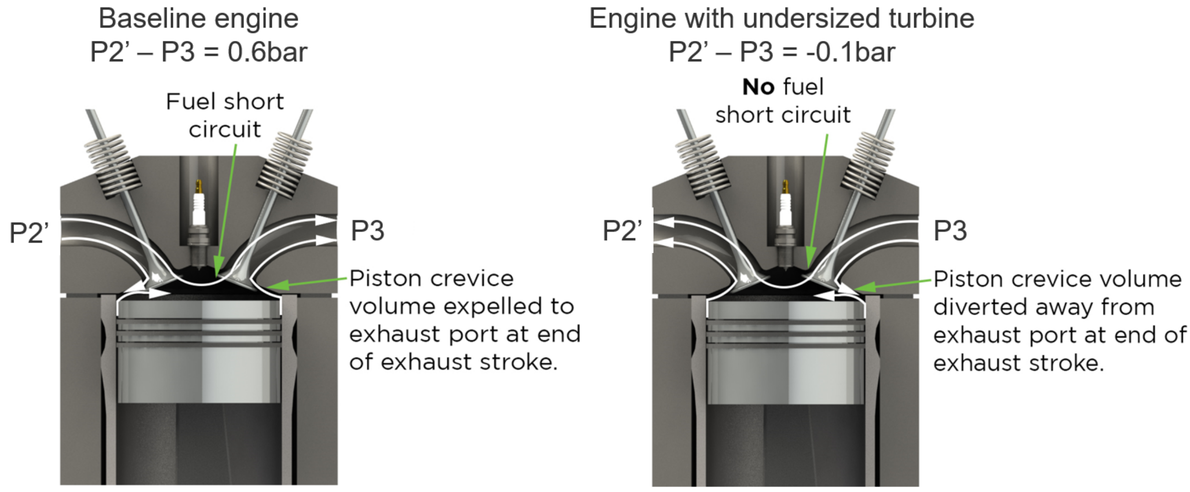

Figure 6: Visualisation of scavenging pressure effect on UHC during valve overlap

2.2.3 Lambda effect on flame quenching

Reducing TC turbine nozzle area together with corrections for constant knock margin or Methane Number (MN) (by retarding spark timing or reducing compression ratio) drive the requirement for lower Air to Fuel Ratios (AFR), or lambda, to achieve the same NOx emissions as the baseline engine (Figure 7). The lower lambda setting results in a further reduction of UHC emissions as a result of improved combustion efficiency. This is due to reduced flame quenching as the flame front stretches and extinguishes as it approaches the cooler cylinder walls and, higher levels of post-oxidation of UHCs during the expansion process and within the exhaust manifold with the higher exhaust temperature (T3).

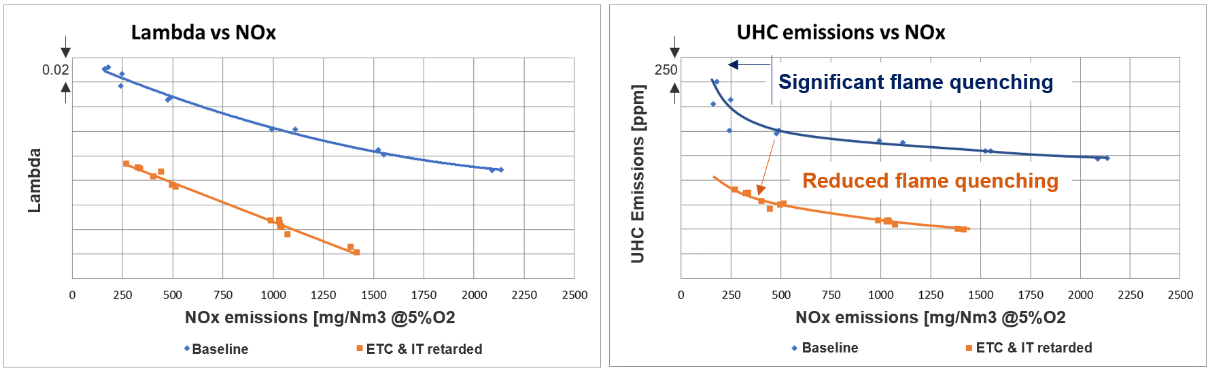

Figure 7: Effect of ETC application on measured lambda and UHC

Figure 7 shows data measured on a lean burn natural gas engine with and without ETC fitted. Fuelling and spark timing were adjusted to achieve equivalent MN capability as the baseline engine at the same NOx emissions. Significant reductions in UHC emissions were measured, with indications that approximately half of the reduction was due to the scavenging effects on fuel slip and half due to the reduced in-cylinder quenching and increased HC post-oxidation effects of running a lower lambda and higher T3.

2.3 ETC and E-Turbo Efficiency walk

Figure 8 shows a typical efficiency walk for a state-of-the-art natural gas genset with ETC applied and the genset adjusted to achieve the same system power and MN capability as the baseline at NOx 250mg/Nm^3 @5%O_2. The typical influence of each of the factors described in sections 2.1 and 2.2 above is quantified.

It is typical to achieve 1.7% points increase in electrical efficiency should the system be optimised at ISO conditions.

The efficiency walk with the same genset modifications, but with E-Turbo applied will typically achieve 1.4% points increase in electrical efficiency at ISO conditions.

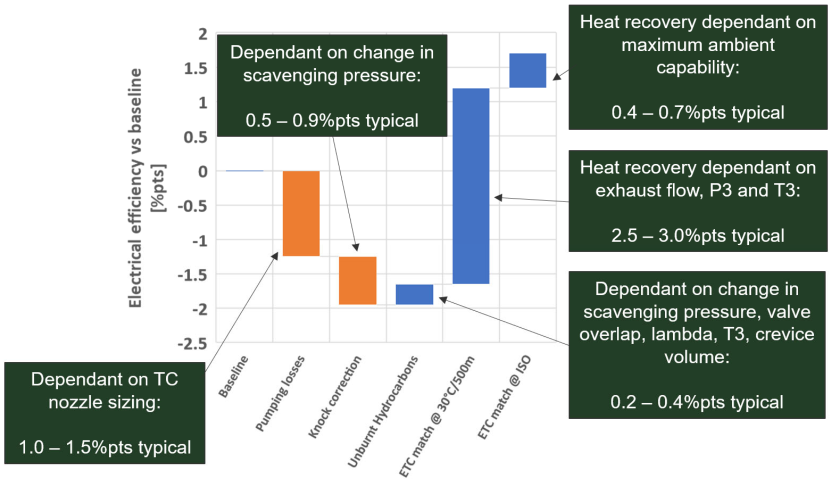

Figure 8: Efficiency walk for ETC applied to lean burn natural gas genset at NOX 250mgNm^3 @5%O_2

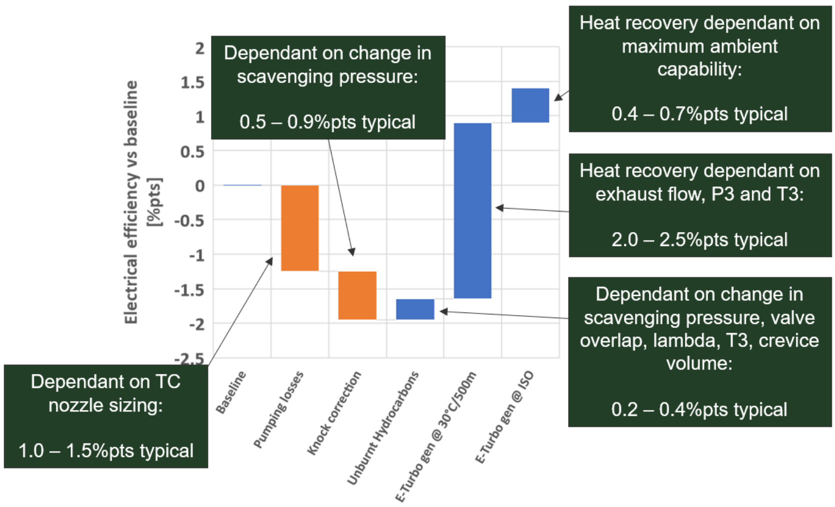

Figure 9: Efficiency walk for E-Turbo applied to lean burn natural gas genset at NOX 250mgNm^3 @5%O_2

3.0 Steady state ETC & E-Turbo simulations

A 1D simulation model was created and correlated to measurements taken on a baseline state-of-the-art natural gas genset running with NOx emissions tuned to 250mg/Nm^3 @ 5% O_2, and the same genset with ETC applied. When applying ETC, the TC nozzle area was optimised while adjusting the spark timing and fuelling to give the same NOx emissions and MN capability at the same system (Genset + ETC) load as the baseline genset.

All results presented were generated using the 1D simulation model and have been corrected back to represent 1MWe at 100% load for simplicity.

The engine governing is achieved using a throttle only. Governing or throttle reserve is calculated by dividing the pressure drop across the throttle by the pressure upstream of the throttle, 100 x (P2 – P2’) / P2.

3.1 Baseline engine considerations

The baseline engine had higher measured UHC emissions versus that measured on other natural gas engines due to a large valve overlap area and increased fuel slip. It is estimated that the influence of applying ETC to this engine on electrical efficiency is approximately 0.2%pts higher than should the engine have had a more typical valve overlap area.

Additionally, all comparisons have been made at NOx emissions 250mg/Nm^3 @5%O_2 in line with recent European market trends. At the more traditional emissions set point of 500mg/Nm^3 @5%O_2 a further reduction in electrical efficiency benefit with ETC of 0.2%pts was measured. This was primarily due to the retarded spark timing and lower lambda set point used having less influence on flame quenching and combustion efficiency at NOx 500mg/Nm^3 @5%O_2 relative to 250mg/Nm^3 @5%O_2.

3.2 Efficiency versus system load

Four turbocharging configurations with HSEM technology have been simulated and the steady state performance compared versus the baseline genset (Figure 10).

Two of the simulation models incorporated ETC together with a TC turbine nozzle available from the TC supplier which had an approximately 20% reduction in effective area with similar turbine efficiency versus the baseline. These models were tuned with two separate ETC turbine nozzle configurations. One sized to give the best system electrical efficiency while achieving the same maximum ambient / altitude capability (or governing reserve) as the baseline genset at 100% load. The other was sized to give best system electrical efficiency at ISO conditions with minimum governing reserve available to run 100% load at steady state.

Two other layouts were simulated with an E-Turbo layout. One with no change to the TC aerodynamic specification and the other with the same ~20% reduction in TC nozzle area as used with the ETC configurations. The power generated by the HSEM for both configurations was increased until the minimum governing reserve for steady state control was achieved at ISO conditions at all loads.

Figure 10a shows that best possible full load electrical efficiency can be achieved using ETC matched at ISO conditions. However, as load is decreased the electrical efficiency benefit decreases. The E-Turbo option with undersized TC nozzle area gives a lower electrical efficiency benefit at 100% load than ETC. However, the added flexibility of being able to adjust the governing reserve using the load generated by the HSEM gives much better part load performance than with ETC. It can be seen that the electrical efficiency benefit with E-Turbo already exceeds that of ETC at 90% load

(Note: The shape of the electrical efficiency benefit curve with E-Turbo is highly dependent on the shape of the TC efficiency curve versus load).

It could therefore be argued that ETC is the best option for base load applications with 100% continuous power operation. In applications where the load profile is biased towards <100% continuous power operation, then E-Turbo would be the preferred solution.

Figure 10b highlights that when using ETC and considering the loss in exhaust energy post TC system (i.e. that recoverable by a heat exchanger for example), a similar total efficiency to the baseline genset at 100% load is achieved. Total efficiency then decreases versus the baseline as the load is decreased. In applications where exhaust energy recovery post-TC system is valuable (for example Combined Heat and Power (CHP) installations) then ETC would only be beneficial should the application be running continuously at 100% power with the electricity price far outweighing the value of the heat.

E-Turbo on the other hand gives an overall increase in total efficiency over the entire load range with only a small loss in exhaust energy post-TC system. It is therefore likely that an E-Turbo solution would be the preferred solution in applications where the exhaust heat is of value such as CHP.

Figure 10c shows that in the cases were the TC nozzle area is reduced, and the engine operation is adjusted to ensure constant MN capability and NOx emissions, a reduction in peak cylinder pressure will be observed at the same system load. This significantly reduces the risk of implementing these technologies from a mechanical or durability point of view. Conversely, these adjustments also result in exhaust temperatures in the order of 30°C higher than baseline. Although modest, it will depend on the baseline engine margins as to whether this will drive a requirement for additional mechanical development work regarding exhaust valve and seats / exhaust system / TC turbine specification etc. It would be possible to avoid the need to do any mechanical engine development by utilising an E-Turbo with the same aerodynamic TC specification as the baseline, however the steady state performance benefits would be much more modest, as indicated in Figure 10a and 10b.

The HSEM sizes required for best performance matched at ISO conditions is ~75kWe for ETC and ~60kWe for E-Turbo (Figure 10d).

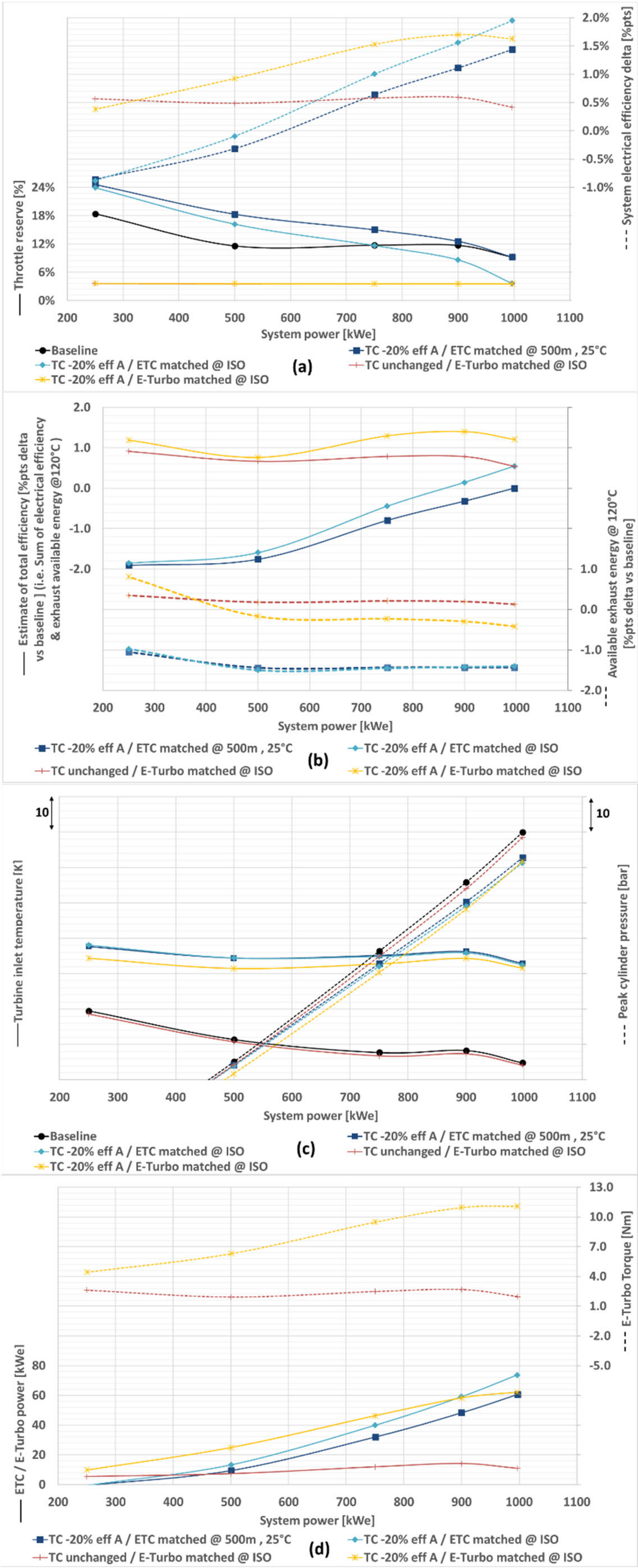

Figure 10: Genset performance and electric machine comparisons for ETC and E-Turbo configurations versus baseline, simulated at ISO conditions

Figure 11 shows the predicted performance of the baseline genset and HSEM turbocharging options versus ambient temperature at 1bar ambient pressure.

It is possible to significantly extend the ambient temperature capability of the genset using HSEM turbocharging options. Figure 11a shows it is possible to extend 100% load operation to well in excess of 45°C without the need to change TC trims. To do this the TGBP valve, in the case of ETC, must be opened (Figure 11d) or the HSEM load generated by the E-Turbo must be decreased (Figure 11b) as ambient temperature increases beyond 25°C. This results in a loss in electrical efficiency of approximately 0.7%pts per 10°C or 0.4%pts per 10°C respectively as the ambient temperature increases beyond 25°C.

Additionally, at lower ambient temperatures, where governing reserve is higher for the baseline genset, it is possible to further improve the electrical efficiency benefit with the E-Turbo options by further increasing the load generated by the HSEM. Considering the average temperature in Europe through the year is ~10°C, the electrical efficiency benefit of the Genset could be increased by an additional 0.5% at these temperatures. This would require the E-Turbo electrical machine to be increased by a further ~13kWe (Figure 11b) relative to being sized for optimal performance at ISO conditions only (Figure 10d). This takes the HSEM size for the E-Turbo to ~75kWe if the cold ambient benefits are to be fully recovered for a 1MWe gas genset (with ~20% reduction in TC nozzle area).

It should be noted from Figure 11c that it is only possible to significantly increase the ambient / altitude capability using HSEM turbocharging options if the TC match of the baseline engine is not speed limited. For example, to run up to 45°C ambient then the TC would need to be capable to run as much as 3 to 5% faster versus the baseline TC at ISO conditions.

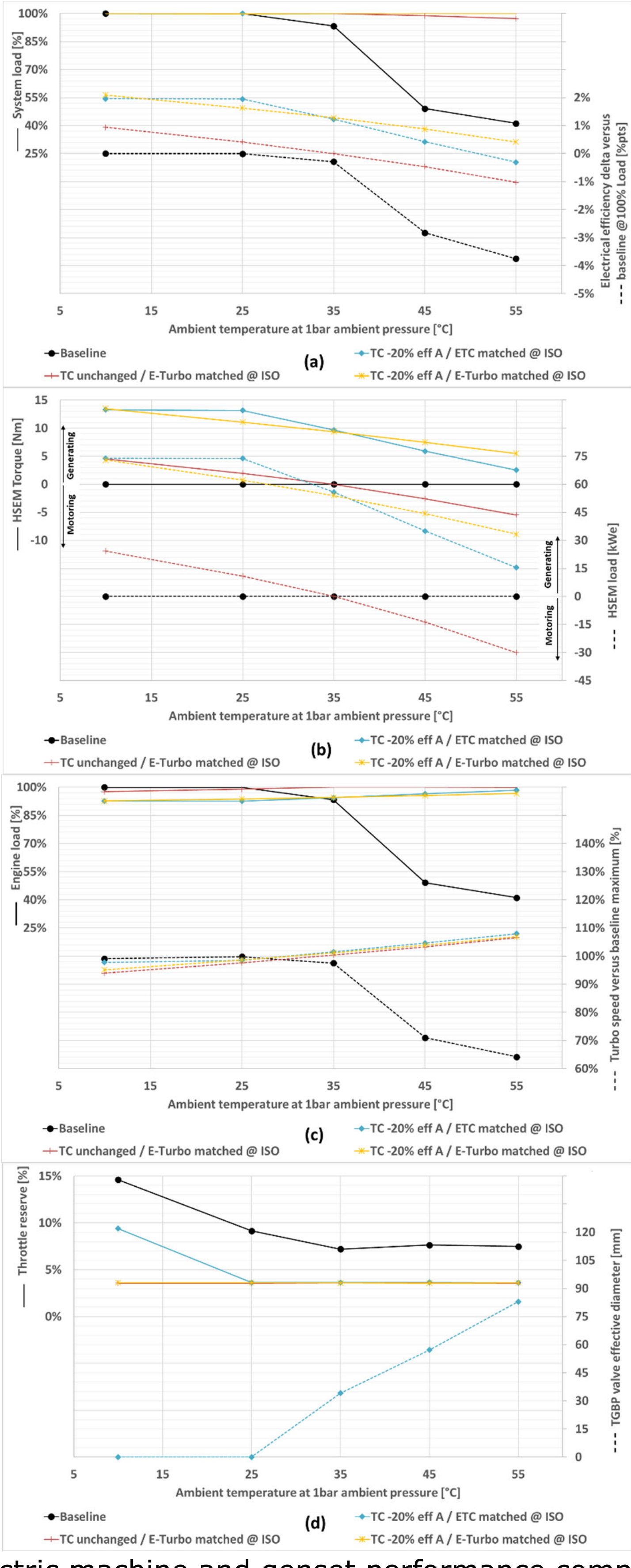

Figure 11: Electric machine and genset performance comparisons versus ambient temperature at 1bar ambient pressure

4.0 Transient performance simulations

4.1 E-Turbo

The ability to motor the E-Turbo using the HSEM provides the potential to significantly enhance transient load acceptance.

In addition, downsizing the TC nozzle area also gives significant transient performance improvement potential, particularly at low loads. For example, if the HSEM is not generating then significant governing reserve, >25% throttle margin can be unlocked to help overcome turbocharger lag and give instantaneous load response (Figure 12). Note: in the case presented a compressor bypass valve would need to be employed to prevent the operating point crossing the compressor surge line at steady state when the HSEM is not generating.

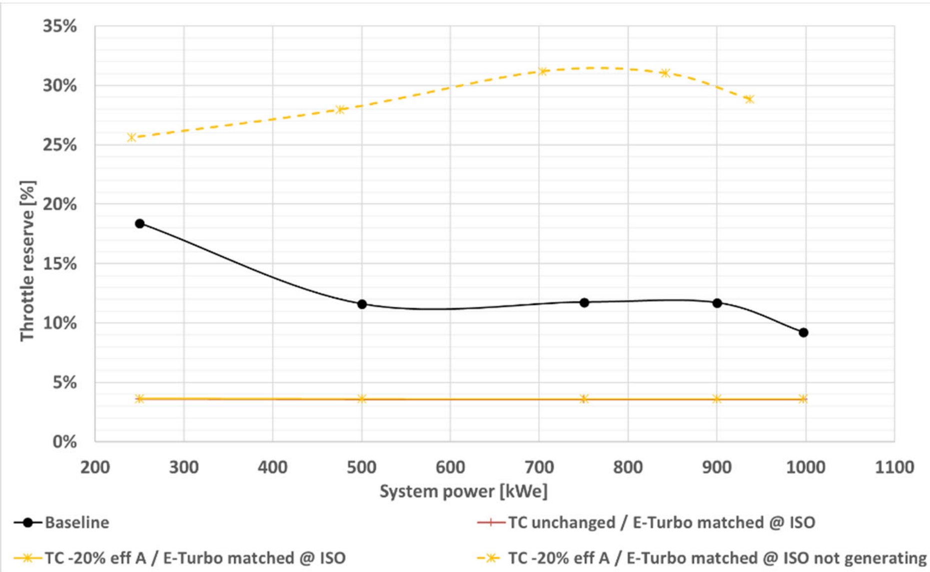

Figure 12: Genset throttle reserve for E-Turbo configurations versus baseline, simulated at ISO conditions

Simulations have been performed to understand the benefits of E-Turbo with and without undersized TC turbine area on pre-heated genset load ramp times, island mode load acceptance, on-grid load ramps and emergency standby start button to 100% acceptance times.

4.1.1 Pre-heated load ramp

As is typical with cold / pre-heated high efficiency gas gensets, significant turbocharger lag during the early part of the load ramp is observed in the baseline simulation, Figure 13. At the time of electrical circuit breaker close (at 10 seconds in the plot) the throttle snaps open giving an initial 10 – 15% load increase. After this, the load increase is stifled as the TC speed increases only very slowly, over many tens of seconds, whilst the power cylinder, exhaust manifold, turbine wheel and turbine housing heat up. This turbo lag continues until the engine and exhaust system components heat up sufficiently that exhaust heat energy reaches and begins to accelerate the TC (approximately 110 to 120seconds after the electrical circuit breaker closes in the baseline simulation).

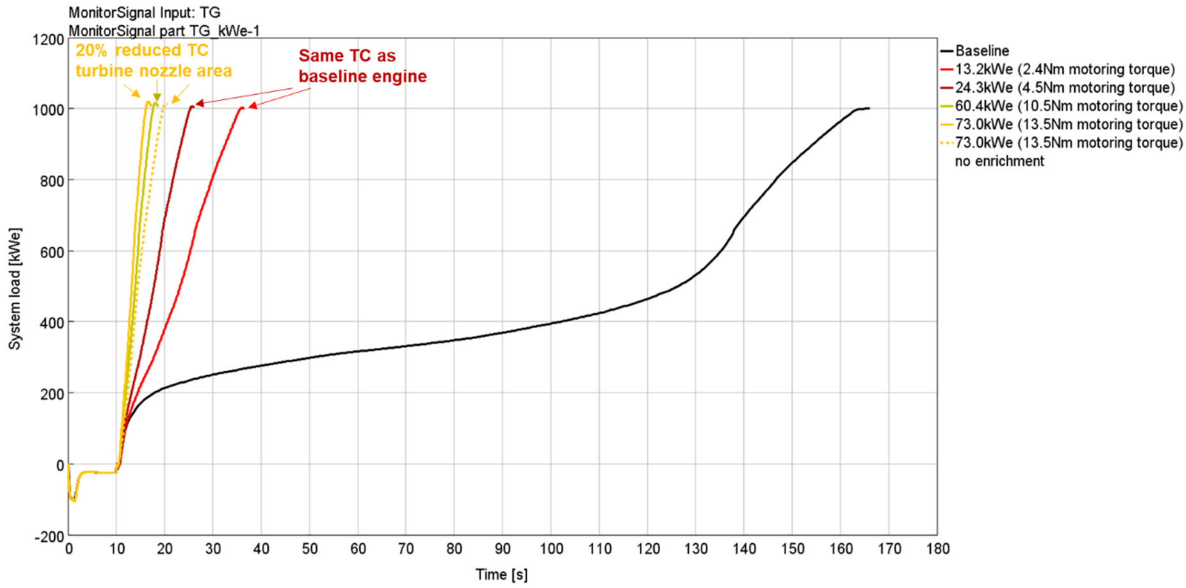

Figure 13: E-Turbo pre-heated genset 0 to 100% load ramp versus baseline

Five scenarios have been simulated to understand the influence of E-Turbo. Two simulations with the TC unchanged and HSEM sized to give optimal heat recovery at ISO conditions (13.2kWe) and again at 1bar/10°C (24.3kWe). Two simulations have been performed with 20% reduction in TC turbine nozzle area and HSEM sized more aggressively to give best heat recovery at ISO (60.4kWe) and 1bar/10°C (73.0kWe). The latter option was additionally simulated without fuel enrichment i.e. using the steady state lambda map for constant NOx emissions.

It comes as no surprise that the ability to motor the TC, even with a small HSEM torque, allows the TC to be accelerated throughout the load ramp overcoming the TC lag. Even with the modest HSEM sizes, with the same TC specification as the baseline, it is possible to decrease load ramp times to 20 to 30 seconds duration. With the more aggressive electric HSEM machines, and modified TC turbine trim, it is possible to reduce the load ramp times to below 10 seconds.

It is even possible to achieve fast start-up load ramp times without fuel enrichment, which should significantly reduce instantaneous NOx emissions and significantly reduce cumulative or total NOx emissions generated and fuel consumed. Additionally the higher lambda set point used will significantly reduce component material temperature gradients during the genset start-up phase, meaning the impact of each engine start / thermal cycle event on the mechanical integrity of the power cylinder and turbocharger components will be reduced versus the baseline, which requires maximum enrichment for approximately 2 minutes.

4.1.2 Island mode

Simulations were performed to assess the impact E-Turbo has on island mode load acceptance capability versus the baseline genset. The layout with 20% undersized TC turbine required a compressor bypass (CBP) valve to be used at low loads to ensure the operating point stayed within the TC compressor map at steady state.

Several basic assumptions were made within the model to ensure a fair comparison for all simulations.

- 80ms to detect the load step, regardless of magnitude.
- 80ms ramp for throttle, CBP, and fuel control valve to saturate once load step detected.
- 20ms torque ramp for E-turbo motoring torque to be applied once load step detected.
- The commands were saturated through the transient with no consideration for smooth recovery.
- Fuelling enrichment used the same lambda limit curve based on intake manifold pressure.
- The same Automatic Voltage Regulator (AVR) settings were used to reduce generator voltage as frequency decreases to aid with the load recovery.
- The E-Turbo was assumed to have 30% more rotational inertia than the baseline.

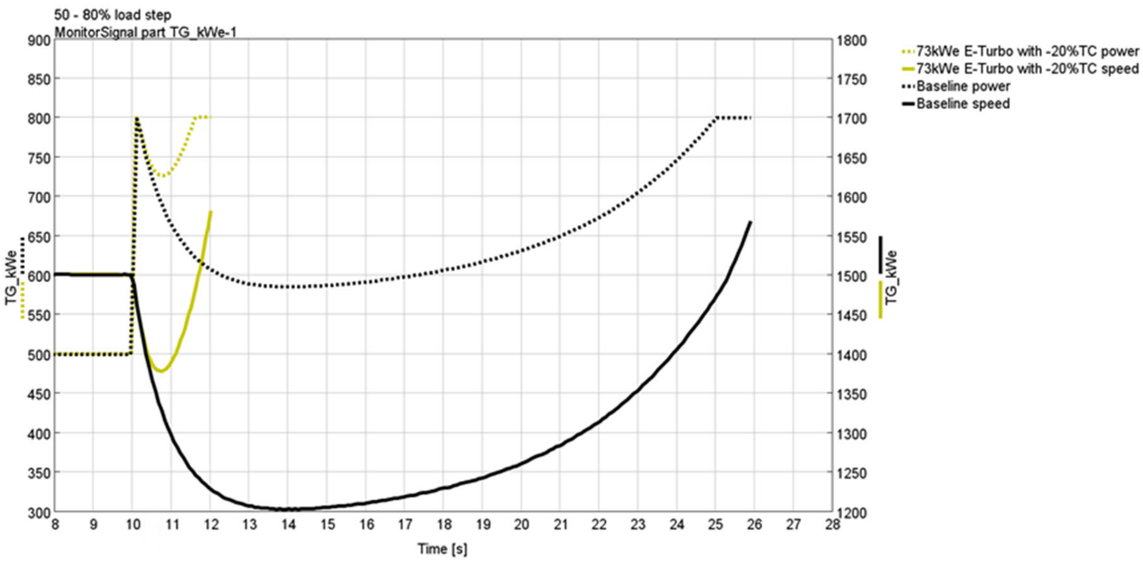

Figure 14: E-Turbo island model load step from 50 to 80% load versus baseline

Figure 14 shows a 50 to 80% instantaneous load step applied to both the baseline genset and E-Turbo layout with undersized TC turbine area and the E-Turbo not generating. The baseline genset almost fails to recover from the load step with the engine rpm dropping to almost 1200rpm and taking approximately 15 seconds to recover. The additional governing reserve available with the E-Turbo layout (Figure 12) means significantly more boost pressure and torque is available within a few engine cycles of the load step being applied. Together with the HSEM motoring the TC, the engine can produce enough boost pressure and torque to overcome the load applied and start the frequency recovery in less than 1 second. With this E-Turbo layout the frequency deviation observed is less than half and recovery time an order of magnitude less than the baseline genset.

The data from this and many more load acceptance simulations carried out at different loads were analysed using the ISO 8528, pt5 criteria [5] to produce Figure 15. Across the full load range, significant improvements in island mode performance were achieved. With the E-Turbo with TC Turbine area 20% reduced and 75kWe HSEM, G3 compliance can be achieved with load steps approximately four times greater than with the baseline genset, providing load acceptance performance comparible to that of a modern diesel genset.

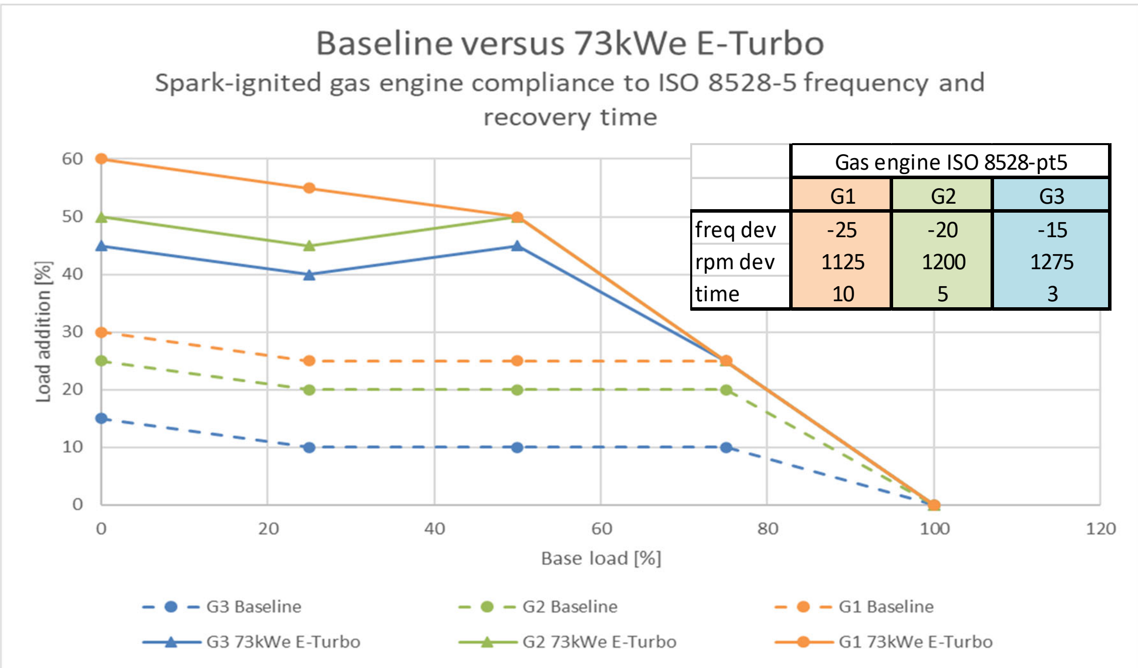

Figure 15: E-Turbo island mode load acceptance comparison versus baseline using ISO 8528-pt5 criteria for turbocharged spark ignited gensets [5]

Similarly, simulations with the E-Turbo layout with the same TC and 24kWe HSEM showed it was possible to achieve G3 compliance with load steps approximately two times greater than with the baseline genset.

4.1.3 On-grid load ramps

Simulations performed for an on-grid scenario showed a step change in load ramp capability can be achieved, Figure 16. The baseline model can be ramped from 25% to full load using agressive fuel enrichment in approximately the same time it takes for the E-Turbo model to be cycled from 25% to 100% to 25% load approximately 3 times without fuel enrichment, avoiding large spikes in NOx emissions in the process.

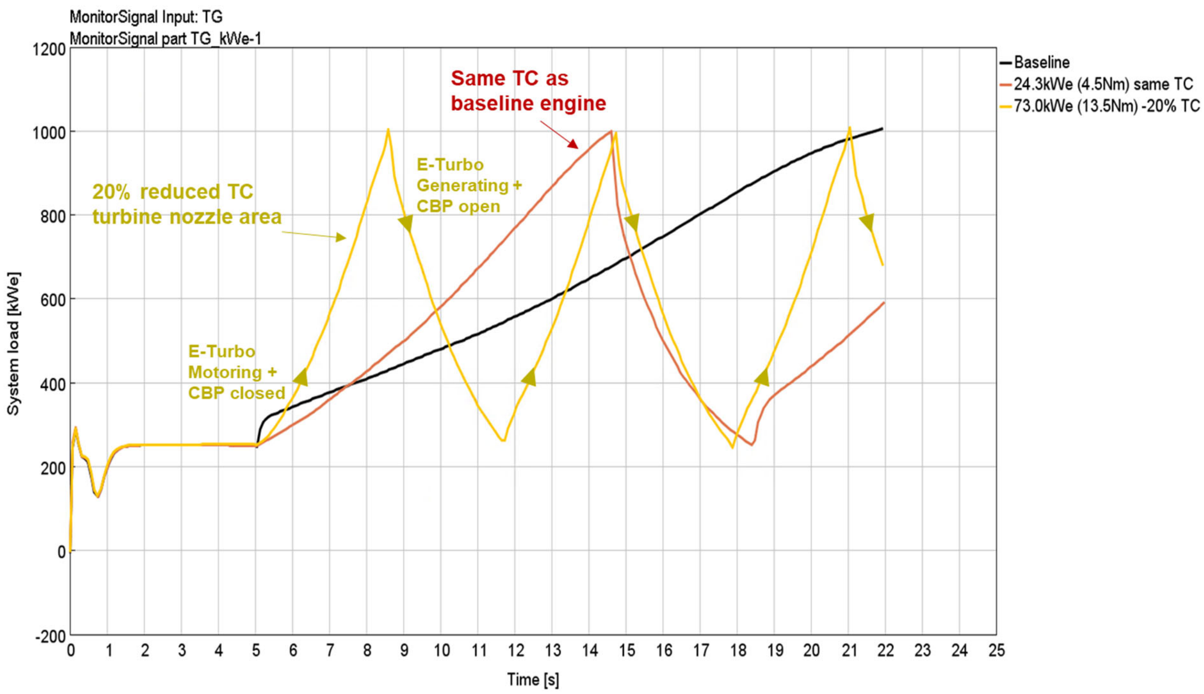

Figure 16: E-Turbo models on grid load ramp capability without fuel enrichment versus baseline with fuel enrichment

4.1.4 Emergency standby

Considering emergency standby scenarios (Figure 17), genset acceleration times could be significantly improved if it is possible to start motoring the HSEM at the time the start button is pressed (at 0 seconds in the plot). If aggressive HSEM accelerations are to be achieved during the speed ramp, then it is necessary to have a CBP valve present and operated to avoid compressor surge while the engine speed is low. The CBP can then be closed as the operating point moves into the heart of the compressor map when the engine speed rises. With this approach, genset speed ramps 2 to 3 times faster than with the baseline can be achieved. Intake manifold pressures in excess of 2 bar can be achieved by the time the engine reaches full speed, allowing the possibility to immediately apply a single, 100% load step to the genset.

It is therefore conceivable that E-Turbo layouts on lean burn gas gensets could even be capable of achieving best in class diesel genset emergency standby performance with the possibility to achieve start button to 100% load acceptance in less than 10 seconds.

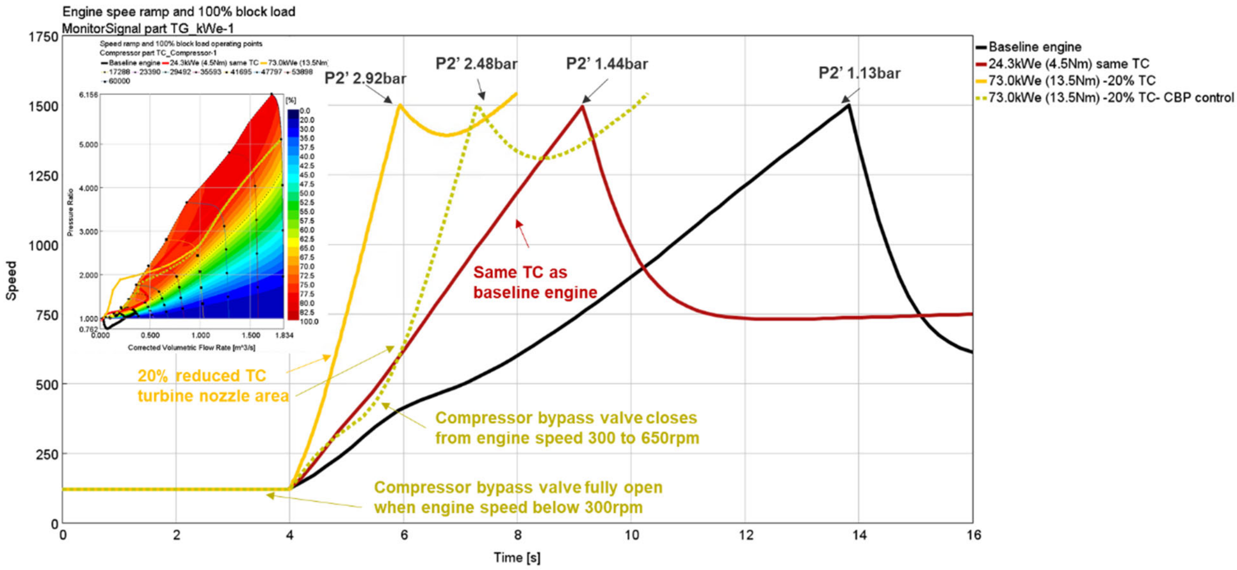

Figure 17: E-Turbo emergency standby speed ramp versus baseline. 100% load step applied immediately when engine speed reaches 1500rpm

4.2 E-Compressors

A final study to understand the effect of using an E-compressor (E-Comp) on the baseline genset pre-heated load ramp time was performed. Three configurations were simulated with an E-Comp integrated in the intake system in series to, or in parallel with the TC compressor, and with the E-Comp integrated into the exhaust system, upstream of the TC turbine, Figure 18.

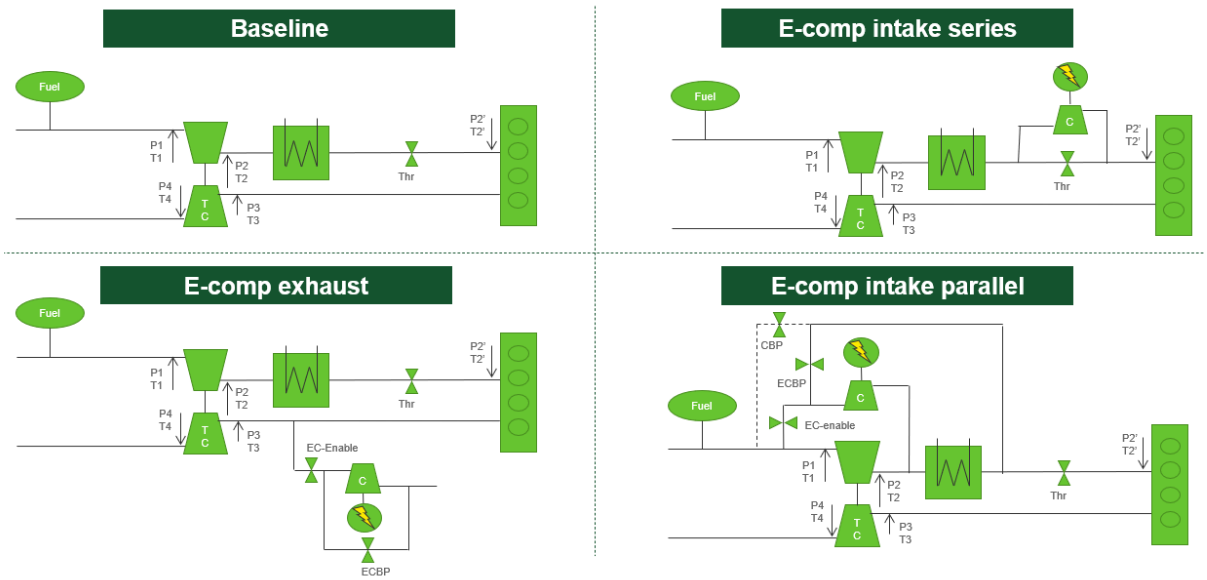

Figure 18: E-Comp layouts considered for pre-heated genset 0 to 100% load ramp

The results in Figure 19 indicate that all configurations are capable to significantly reduce the genset load ramp time. Each configuration has their own challenges in terms of controllability, sizing, packaging, and complexity, however.

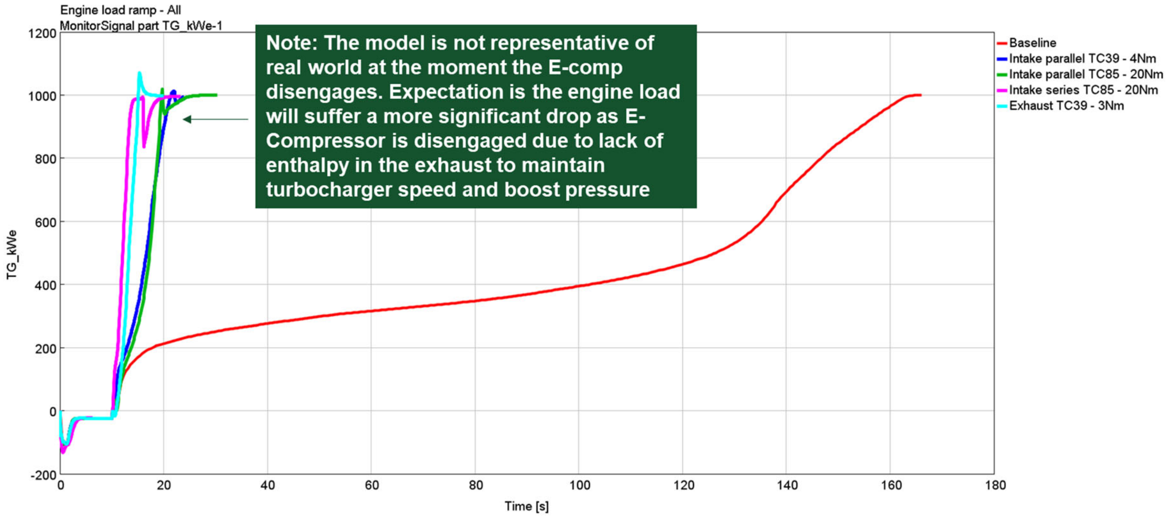

Figure 19: Pre-heated genset 0 to 100% simulations. E-Comp motored at time of breaker closing (at 10 seconds) for the intake system layouts. E-Comp motored prior to breaker closed for exhaust system layout (at 0 seconds).

From a controls perspective, the intake series layout is the simplest with little concern of TC or E-Comp compressor surge and thus no need for additional actuators on the engine. As all the flow going to the engine must pass through the E-Comp compressor, it needs to be of similar size to that of the TC compressor. This drives the requirement for a large HSEM to have a desirable impact. This presents significant packaging and cost challenges when considering its addition to an existing engine platform.

The amount of additional air that can be compressed by the E-Comp in the intake parallel solution is limited by the operating point on the TC compressor map and its surge line. This gives a limit to the E-Comp compressor and HSEM sizes that can be used. Trying to increase the E-Comp size will only result in the need to operate a CBP valve around the TC compressor to avoid surge, resulting in no possibility to further improve the load ramp time (See 4 and 20Nm plots in Figure 19). This does limit the size and cost of the E-Comp to a reasonable price range, however two additional valves still need to be used, adding complexity and controls challenges when trying to avoid compressor surge on both TC and E-Comp compressors during the load ramp.

The E-Comp exhaust solution can have a significant impact on the load ramp of the genset with a relatively small E-Comp without affecting the TC compressor operating line and therefore with little impact on the base engine layout or controls. However, an additional valve is needed to be able to isolate the E-Comp from the exhaust and avoid exhaust leakage when not in use, and a further valve is needed to bypass the E-Comp compressor to avoid surging towards the end of the load ramp. This bypass does provide the ability to motor the E-Comp HSEM to full operating speed prior to the genset breaker closing, giving an increased TC acceleration at the start of the load ramp.

The three E-comp layouts have been compared in Table 1 with the E-Comp exhaust solution giving the best overall opportunity for improving genset load ramp times with minimal cost and impact to the base engine. This solution does not need to be matched to a specific engine and could potentially be used without modification across many engine types and platforms.

Table 1: E-Turbo selection matrix for 1MWe gas genset

+ / - Indicative relative to each other	E-Comp intake parallel (~4Nm / 30kWe)	E-Comp intake parallel (~20Nm / 100kWe)	E-Comp intake series (~20Nm / 100kWe)	E-Comp exhaust (~3Nm / 20kWe)
Start-up impact	-	-	+	+
Size / Cost	0	-	-	+
Integration complexity	0	0	-	+
Controls complexity	-	-	+	0
Exhaust scavenging potential	-	-	-	+
Applicability to multiple genset platforms	0	0	-	+
Total	**- 3**	**- 4**	**- 2**	**+ 5**

5.0 Summary

The outputs from the various steady state and transient studies described in sections 3 and 4 have been summarised and compared in Table 2.

The steady state electrical and total efficiency benefits simulated and plotted in section 3 with the TC modified with 20% reduced nozzle area, were reduced by 0.2%pts for the reasons described in section 3.1

Table 2: E-Turbomachinery comparison for typical 1MWe gas genset at NOx 250mg/Nm^3 @ 5%O_2, same system load and same MN capability

Indicative relative to Genset at 100% system load		Gas Genset (Baseline)	ETC + TGBP	E-Comp (Exhaust)	E-Turbo	E-Turbo
Turbocharger modification		-	~20% reduction in nozzle area	None	None	~20% reduction in nozzle area
HSEM size		-	75kWe	20kWe	25kWe	75kWe
Efficiency	**Electrical efficiency benefit @ ISO**	-	+ 1.7%pts	-	+ 0.4%pts	+ 1.4%pts
	Electrical efficiency benefit @ 1bar 10°C	-	+ 1.7%pts	-	+ 0.9%pts	+ 1.9%pts
	Electrical efficiency benefit @ 1bar 55°C (System load achieved @ 1bar 55°C)	-3.8%pts (41%)	-0.5%pts (100%)	-3.8%pts (41%)	-1.0%pts, (97%)	+0.2%pts, (100%)
	Total (electric + exhaust) efficiency benefit & ISO	-	+ 0.3%pts	-	+ 0.4%pts	+ 1.0%pts
Emissions reduction (kg/MWehr)	**Unburnt hydrocarbon**	-	- 20 to 40%	-	- 4%	- 20 to 40%
	NOx	-	- 8%	-	- 4%	- 8%
Transient operation	**Grid balancing**	-	Low load benefit*	Low load benefit*	2x ramp rate	5 x ramp rate
	Island mode (ISO 8528-pt5)	G3 @ 10% step	G3 @ 20% step	Low load benefit*	G3 @ 20% step	G3 @ 40% step
	Pre-heated start load ramp time (s)	150	70*	5	15	6
	NFPA 110 / emergency standby potential	No	No	No	No	Yes
Applicability	**Applicability to multiple engine platforms**	-	+	+	-	-
	Applicability to other markets e.g. rail, marine	-	-	-	+	+

* Expectation - Not measured or simulated as part of this study.

6.0 Conclusion

BPG's experience of designing and applying HSEM technology to turbomachinery has been used to understand and report the achievable steady state and transient benefits when applying ETC, E-Turbo and E-Comp to a modern 1MWe high speed natural gas genset:

- ETC provides the highest achievable electrical efficiency gain through waste heat recovery at 100% load.
- E-Turbo provides increased electrical and total efficiency through waste heat recovery together with diesel like transients and enhanced controls flexibility.
- E-Comp in the exhaust manifold, pre-turbine enables fast start capability without changes to engine steady state performance, boundaries, or architecture.

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