2010-06-01

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Recommended Citation

Kierans, R., McGovern, J.: Investigation into the Possible Benefits of a Refrigeration Based Intercooling System. 4th International Conference on Sustainable Energy & Environmental Protection, SEEP2010, Bari, Italy. June 29th – July 2nd, 2010.

Investigation into the Possible Benefits of a Refrigeration Based Intercooling System

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Abstract: This paper describes findings from analysis and simulation in relation to the possible benefits of applying refrigeration based intercooling to the engine of a modern passenger car or haulage vehicle. Initial theoretical analysis showed that, depending on the engine specifications and the boost pressure level of the turbo/supercharger, the power improvement could range from 15–30% and there would be a similar improvement in torque. This meant a smaller engine could be used to do the work of a larger engine. The power required to run the refrigeration system, the space required and the charge air pressure drop due to its passage through the system were all within what were believed to be acceptable limits. It was envisaged that the proposed system would have a secondary function as an air conditioning unit to save space. An engine simulation software package, Ricardo WAVE, was used to simulate the engines that had been investigated in the theoretical calculations. The engines were simulated with and without refrigerated intercooling. There was good correlation between the simulation results and the simpler theoretical analyses. Slight gains in efficiency were predicted, but the main potential benefits were increased power and torque output.

Keywords: refrigerated intercooling, engine simulation, performance analysis

1. Introduction

This project was an endeavour to research the possible benefits of using a refrigeration based intercooler system in conjunction with a turbo/supercharged diesel or petrol engine in a car or a truck. It was hoped that such a system would give an increase in power and improve efficiency in an engine over what was currently achieved with a standard intercooler.

Compared to standard intercooling, refrigerated intercooling further increases the density of the charge induced into the cylinder of the engine and so allows more fuel to be burned per cycle, increasing power output. The higher energy release per charge also increases torque output. Better intercooling, through the use of refrigeration, improves the overall air compression process by reducing the compression work per unit mass and would therefore have been expected to yield some improvement in brake thermal efficiency.

2. Literature review

Until this project was undertaken, it appears that little research had been done on this concept with regard to petrol and diesel engines. One similar system which was developed by J.C. Bamford Excavators [1] used a refrigeration system to cool air, which was then used to cool the charge air (air received into the intake of an engine) in the intercooler. A possible disadvantage of this system would have been the series of heat transfers that would have been required. The system proposed in the present paper would use a refrigeration system custom designed to achieve an intercooler output temperature of 4°C, which could not be achieved as economically by using cold air, cooled by a refrigeration system, to cool the air in the intercooler. A patent application was published by Richard H. Klem [2] concerning such an idea in 2002, but, the application only seemed to cover the concept and did not cover how the system operated. The inventor speculated that with such a system the power output could be two or three times as much as in a naturally aspirated engine.

Ford Global Technologies [3] have developed a system similar to the one outlined in this paper for use in a hydrogen combustion development project. So, although there has been some research and development in this area, it may not have been applied to production vehicle petrol and diesel engines.

3. Design

Before any of the calculations [4] could be undertaken in the design of the refrigerated intercooling systems, decisions had to be made in regard to the general nature and functions of the systems.

3.1 Engines chosen

3.1.1 Mercedes C230 Kompressor petrol

The M271 engine installed in the current Mercedes C230 was chosen as the passenger car engine to be analysed. It was chosen because it was a mid-sized supercharged engine with a moderate level of boost pressure. Data for the engine was acquired from motor distributors [7].

3.1.2 Mercedes Actros 3355s V8 diesel

The OM502LA engine installed in the current Actros truck was chosen as the truck engine to be analysed. This engine was ideal for this project as it was a large and very powerful engine with a high boost pressure. Giving this engine a higher power and torque output without a huge weight gain would enable haulage companies to transport larger payloads with the same engine. Data for the engine was acquired from motor distributors [8].

3.2 Refrigeration system

Certain parameters had to be defined before the system could be designed. The first and probably most important decision was the choice of the refrigerant to be used. R134a was selected due to the suitability of its properties and characteristics and the ready availability of information about it. As the required output temperature of charge air from the evaporator was 4°C, the evaporating temperature was set at −4°C so that there would be an 8 K difference between the two temperatures to allow the air to be cooled successfully to 4°C. Similarly, the condensing temperature was set at 30°C so that it would be 10 K above an assumed ambient air temperature of 20°C. Also, 8 K of suction line superheat was specified.

3.3 Standard automotive air-conditioning components

One of the initial concerns was that the components required to meet the cooling loads might be so much larger and require so much extra power, when compared to standard automotive air-conditioning components, that installing a refrigeration based intercooling system would not be practical. However, it was learned that air-conditioning systems in cars could meet cooling loads of over 12 kW [5] at peak, which was higher than the required load for a refrigerated intercooler for the car, as can be seen in the Calculations section. It was also learned that that the Volvo 9700 bus which had a 339 kW, 12 litre, 6 cylinder engine had a 32 kW air-conditioning system [6]. It can be seen in the Calculations section that this air-conditioning load of the bus was not a lot less than that required for the Actros truck, which as standard had an engine with a greater output. This suggested that the required refrigeration load for the refrigerated intercooler in the truck was also achievable.

3.4 Compressor

As the most common type of compressor used in automotive air conditioning systems was the pulley driven scroll compressor, it was decided that this would be the most suitable choice for the compressor to be used in this system. An isentropic efficiency of 70% was assumed, based on the mechanical shaft power input.

3.5 Expansion device

As the system was designed with changeable conditions in mind and because suction line superheat was required, a simple expansion device such as a capillary tube was deemed unsuitable. An expansion device was required that could automatically adjust for changing ambient and operating conditions and be able to assure adequate suction line superheat to protect the compressor from liquid slugging. For these reasons a thermostatic expansion valve was envisaged for this system.

3.6 Condenser

The condenser was designed as a finned tube heat exchanger, as this was the most common type of automobile condenser. One key constraint when designing the condenser was that it had to have dimensions as similar as possible to those of the condensers already being used in conjunction with the two chosen engines. Based on these dimensions the number of tubes and dimensions of the fins were approximated and applied to the design specification of the proposed condensers.

3.7 Evaporator

The evaporator was also designed as a finned tube heat exchanger, as this was the most common type used in automobile air conditioning systems. The shapes and sizes of the proposed evaporators differed greatly from those of the condensers, as the evaporators had to fit into as small a space as possible in each case. Also, the higher density of the refrigerant, the lower heat load on the evaporator and allowing the refrigerant to have a higher velocity meant that the evaporators could be made much smaller than the condensers.

3.8 Dual function

It was decided that to save space and energy the refrigeration system should have a dual function in that it would be used when required to operate the refrigerated intercooler, but, would also provide cabin air conditioning. Maximum load would only be required by the intercooling system when the engine was at maximum speed. The vehicles' engine management systems could be used to perform the task of operating valves to regulate or switch the flow of refrigerant between the evaporators of the two systems.

4 Calculations

Calculations were required to determine the increased power and torque outputs of the engines, the refrigeration loads, the specifications of the refrigeration systems, and the sizes of the heat exchangers.

4.1 Otto cycle calculations

Otto cycle calculations were used to determine the theoretical increases in power for both the petrol passenger car engine and the truck engine. These calculations showed power and torque increases for the C230 engine of 16.28% and 16.39% respectively. The power curve from the calculations for the standard engine was mapped to the actual engine power curve, as provided in the engine data. This mapping was then applied to the second power curve (with refrigerated intercooling) to give it its expected shape. This had to be done as the theoretical calculations gave a straight line and not a curve because they did not take into account pumping and other losses. The power curves, with and without refrigerated intercooling, are shown in Figure 1.

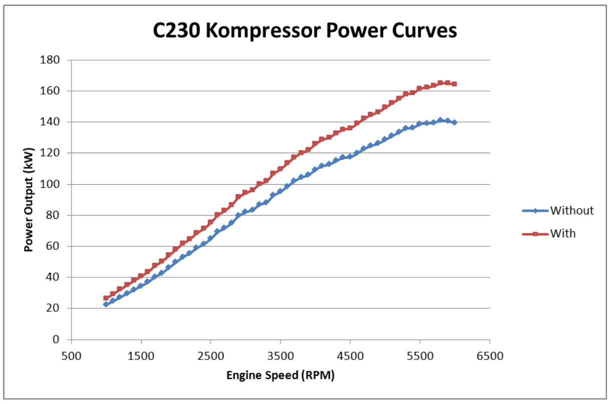

Figure 1. C230 engine power curves

Otto cycle calculations were used for the truck engine because modern diesel engines were more accurately represented by this cycle. The calculated increases in power and torque output for the Actros engine were 28.99% and 28.98%. The power curves are shown in Figure 2. The same method of mapping that was applied to the C230 power curves was also applied to the Actros' power curves.

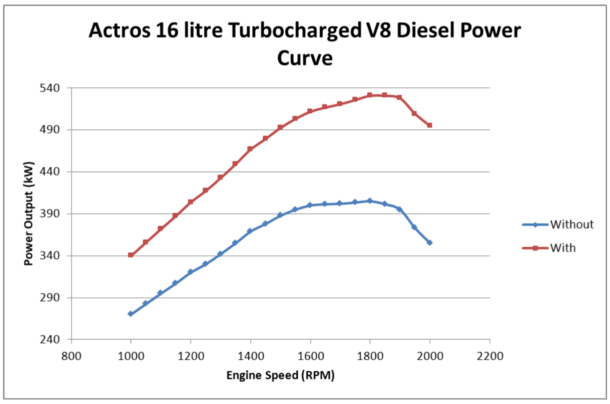

Figure 2. Actros engine power curves

Owing to complications with setting-up the simulation of the Actros engine, which meant that the results produced were not accurate, these results could not be used for comparison to the theoretical results. For this reason a sample truck engine that was provided with the Ricardo WAVE program was used instead. In order for this model to be used, its specifications were input into the theoretical calculations so that there would be theoretical results to which the simulated results for the sample truck engine could be compared. The power and torque in the case of this engine were both calculated to increase by 26.11%. The power curves are shown in Figure 3. The method of mapping calculated results onto the know power curve of an engine was also applied to this virtual engine so that the power curve that resulted from the mapping of the theoretical power curve for the standard sample truck engine replicated the simulated standard power curve. The same mapping was used to approximate the refrigerated power curve based on the theoretical calculations.

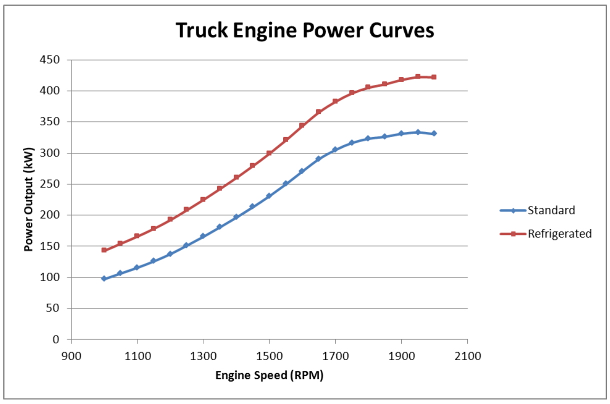

Figure 3. Sample truck engine power curves

4.2 Refrigeration load

The refrigeration load was calculated using a steady flow energy equation. The refrigeration load required for the evaporator for the C230 refrigerated intercooler was 6.67 kW. The refrigeration load in the evaporator required for the refrigerated intercooling system in the Actros was 36.8 kW.

4.3 Refrigeration system

The refrigeration system was designed using reversed Rankine cycle calculations. The proposed refrigeration cycle had a refrigeration COP, based on the shaft power input, of 4.69. The refrigeration effect was 161.5 kJ/kg and the heat rejection effect was 185.6 kJ/kg.

4.4 Heat exchanger sizing

Heat transfer correlations were used to determine the overall heat transfer coefficients of the heat exchangers designed as part of this project. Hence these heat exchangers were sized so that they could deal with the calculated heat load. The overall heat transfer coefficients for the condenser and the evaporator in the C230 were determined to be 1274 W/m^2K and 1639 W/m^2K respectively. The required sizes for the heat exchangers were calculated to be $498 \times 420 \times 16$ mm and $111 \times 200 \times 24$ mm. The overall heat transfer coefficients for the condenser and the evaporator in the

Actros truck were determined to be 1281 W/m^2K and 1856 W/m^2K respectively. The sizes for these heat exchangers were calculated to be 587 × 460 × 80 mm and 104 × 240 × 96 mm.

4.5 Charge air pressure drop

The pressure drop experienced by the charge air as it passed through the evaporator was determined by applying the principles of fluid dynamics for internal flow. The pressure drop through the evaporator required for the C230 was calculated to be 0.052 kPa. The pressure drop through the evaporator required for the Actros was calculated to be 0.457 kPa.

5. Simulated performance results

On reviewing the simulated results for the C230 engine and the sample truck engine it became clear that the performance benefits of a refrigeration based intercooling system in terms of increases in brake power output and brake torque output were considerable. The power gains were as much as 10.59% and 21.37% for the respective engines. These results showed that the proposed system could provide considerable power gains for the engines and therefore should also be able to provide considerable power gains for any engine which was either turbo or supercharged. The power curves that resulted from the simulation of these engines are shown in Figures 4 and 5.

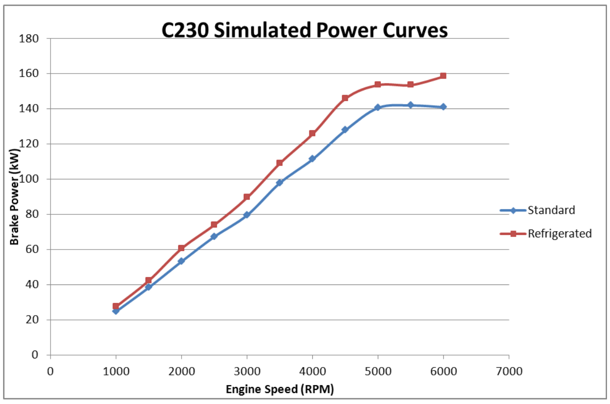

Figure 4. C230 engine simulated power curves

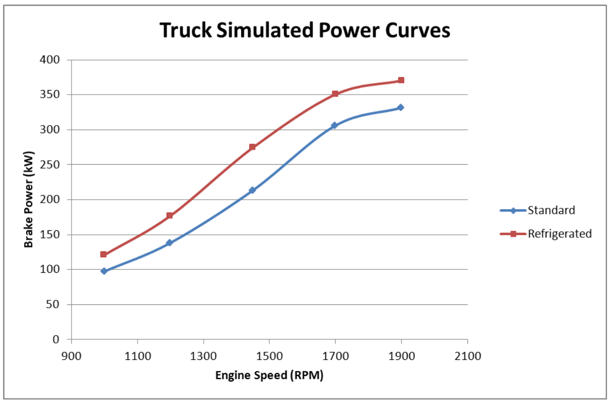

Figure 5. Sample truck engine simulated power curves

The engine models that were successfully simulated showed modest gains in brake thermal efficiency of up to 1.15% and 2.37% respectively. The simulated brake thermal efficiency results for the engines are graphed against engine speed in Figures 6 and 7.

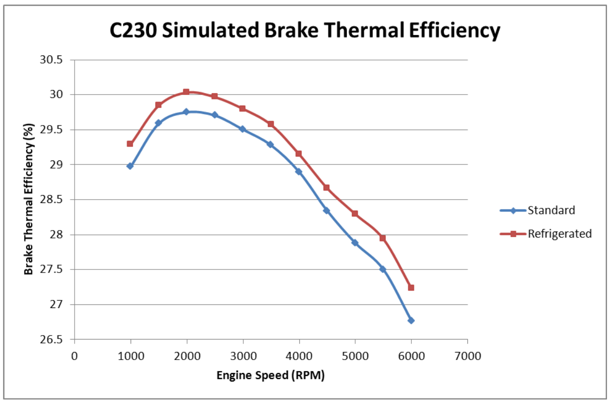

Figure 6. C230 engine simulated brake thermal efficiency

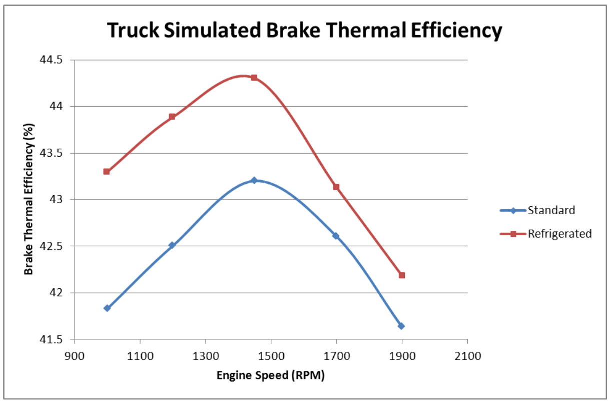

Figure 7. Sample truck engine simulated brake thermal efficiency

The increases in fuel required by the engines were of up to 10.25% and 20.6% respectively and as these increases were proportionately less than the increases in power, there were small reductions in brake specific fuel consumption (corresponding to the small increases in brake thermal efficiency). The results for the simulated fuel volume flow in the two engines are graphed against engine speed in Figures 8 and 9.

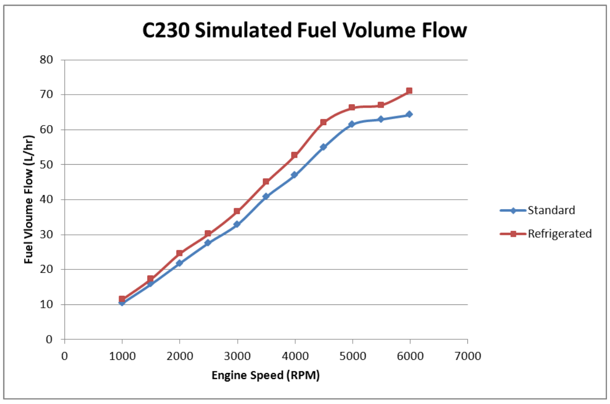

Figure 8. C230 engine simulated fuel volume flow

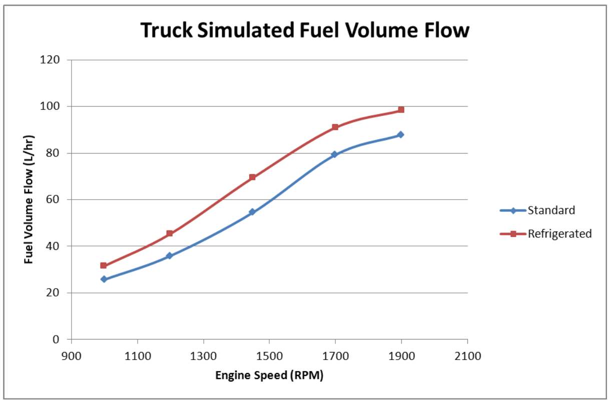

Figure 9. Sample truck engine simulated fuel volume flow

The reduction in exhaust gas temperature would have required to be quite small if the assumption that this temperature remained the same was to be considered a valid assumption. Such an assumption was made in the theoretical analysis so that the increases in brake power and brake torque could be calculated. Simulation showed the reductions in the exhaust gas temperature to be up to 1.24% and 1.6% in the respective engines. These reductions were considered to be small enough changes from the original temperatures to justify the assumption. The simulated results for the exhaust gas temperatures of the engines are graphed against engine speed in Figures 10 and 11.

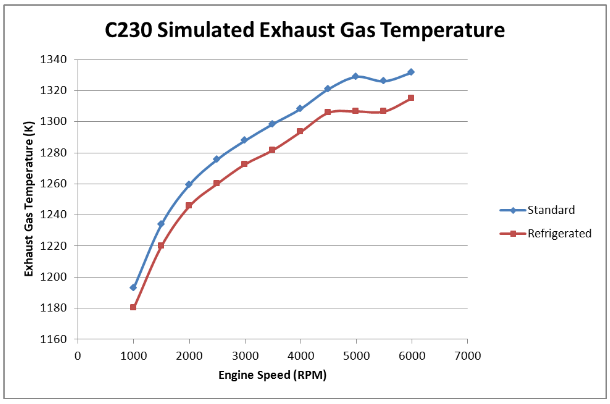

Figure 10. C230 engine simulated exhaust gas temperature

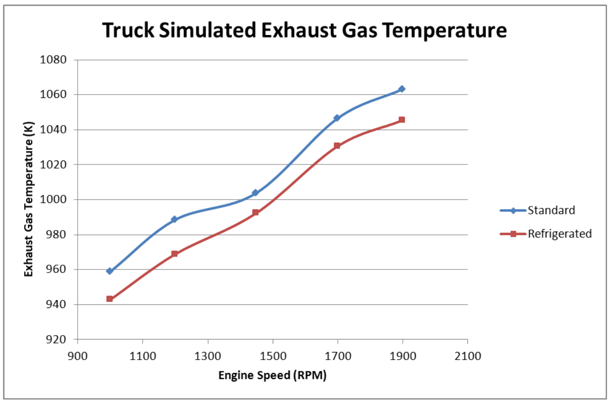

Figure 11. Sample truck engine simulated exhaust gas temperature

6. Simulated versus theoretical results

6.1 C230 engine results

The theoretical and simulated results were compared to each other so that the accuracy of the calculations could be determined. Firstly the results for the C230 M271 1.8 litre four cylinder engine were compared and contrasted. Beginning with the increase in brake power output, the theoretical increase was calculated to be 16.28% and the simulated increase was found to be 10.59%. The difference in these two figures was believed to be due to losses which were not taken into account by the Otto cycle calculations and, to a lesser degree, could also be attributed to the assumption that the combustion temperature would remain the same, as when this temperature was adjusted the calculated performance gain dropped to 14.44%. In terms of the power curves themselves the deviation between the theoretical and simulated power curves was found to be 5.68% for the power curve of the standard engine and 3.64% for the power curve of the engine when the refrigerated intercooler was installed. These deviations changed to 5.68% and 3.96% for the same power curves when the reduction in combustion temperature was taken into account. These deviations were believed to be due to lack of exact correspondence between the engine simulation model and the engine that was analysed theoretically. The engine simulation model was based on a sample engine provided as part of the Ricardo WAVE package and was not a rigorous representation of the actual engine in the C230 vehicle. The theoretical and simulated power curves for the C230 engine are shown in Figure 12. The theoretical and simulated power curves with the adjusted theoretical refrigerated power curve are shown in Figure 13.

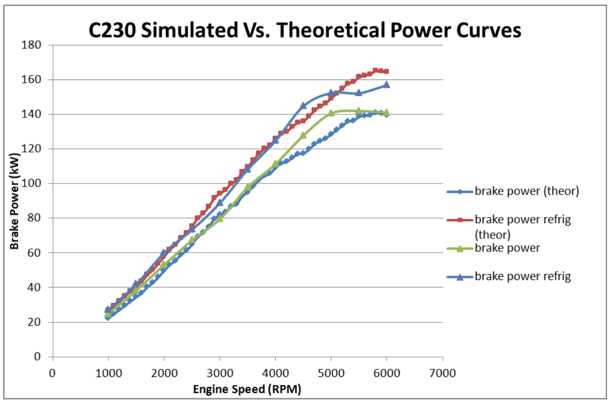

Figure 12. C230 engine simulated vs. theoretical power curves

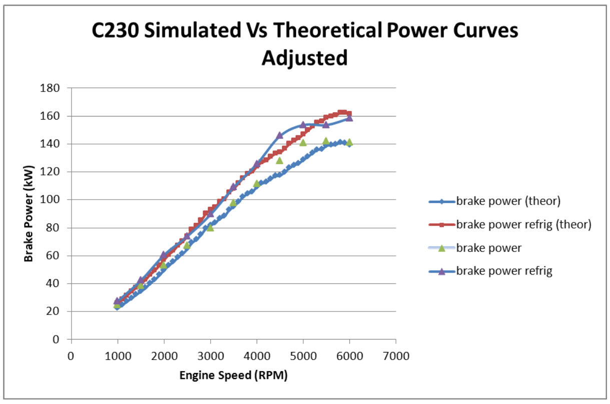

Figure 13. C230 engine adjusted simulated vs. theoretical power curves

In terms of the increase in brake torque output, the theoretical increase was calculated to be 16.39% and the simulated increase was found to be 10.59%. The difference in these two figures was believed again to be due to losses which were not taken into account by the Otto cycle calculations and the assumption that the combustion temperature would remain the same. When this temperature was adjusted the calculated performance gain dropped to 14.51%. In terms of the torque curves themselves the deviation between the theoretical and simulated power curves was found to be 6.89% for the torque curve of the standard engine and 4.89% for the torque curve of the engine when the refrigerated intercooler was installed. These deviations changed to 6.89% and 4.78% for the same power curves when the reduction in combustion temperature was taken into account. These deviations were believed to be due to the fact that, as with the brake power, the engine model used was not solely based on the actual C230 engine. Also it was noted that when the torque curve for the engine with the refrigerated intercooler was adjusted for the decrease in combustion temperature the percentage deviation reduced. The theoretical and simulated torque curves for the C230 engine are shown in Figure 14. The theoretical and simulated torque curves with the adjusted theoretical refrigerated torque curve are shown in Figure 15.

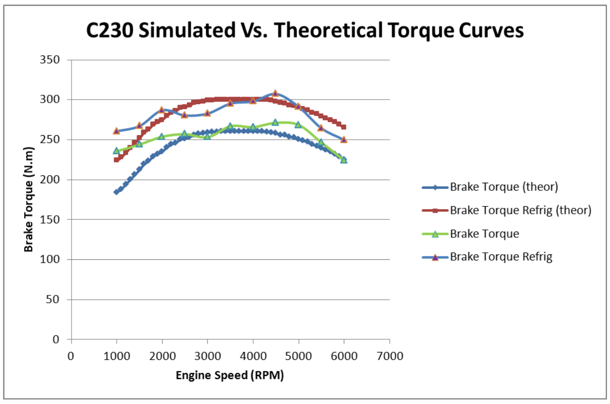

Figure 14. C230 engine simulated vs. theoretical torque curves

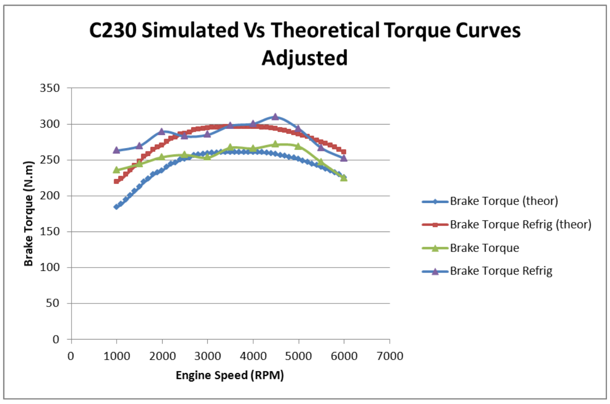

Figure 15. C230 engine adjusted simulated vs. theoretical torque curve

6.2 Sample truck engine results

The results for the same parameters of the 10 litre six cylinder truck engine were then compared and contrasted. Beginning again with the increase in brake power output, the theoretical increase was calculated to be 26.1% and the simulated increase was found to be 21.37%. The difference in these two figures was believed to be due to losses which were not taken into account by the Otto cycle calculations and to a lesser degree could also be attributed to the assumption that the combustion temperature would remain the same. When this temperature was adjusted, the calculated performance gain dropped to 22.75%. In terms of the power curves themselves the deviation between the theoretical and simulated power curves was found to be 0.24% for the power curve of the standard engine, 8.91% for the power curve of the engine when the refrigerated intercooler was installed and 6.5% for the same power curve when the reduction in combustion temperature was taken into account. The deviation for the standard power curve was quite small, as was expected, since the theoretical curve was mapped to model the simulated curve, which was regarded as providing an accurate representation of the actual power for this engine. The deviation of the curve when the refrigerated intercooler was installed was believed to be due to losses not taken into account by the Otto cycle calculations. The fact that the deviation reduced when the drop in combustion temperature was taken into account was believed to be because this reduced the number of losses not taken into account by the Otto cycle calculations. The theoretical and simulated power curves for the sample truck engine are shown in Figure 16. The theoretical and simulated power curves with the adjusted theoretical refrigerated power curve are shown in Figure 17.

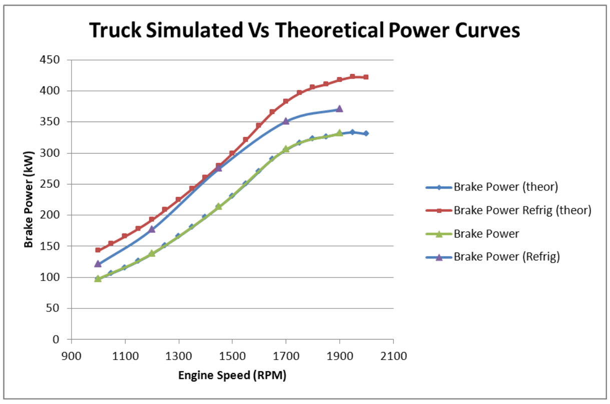

Figure 16. Sample truck engine simulated vs. theoretical power curves

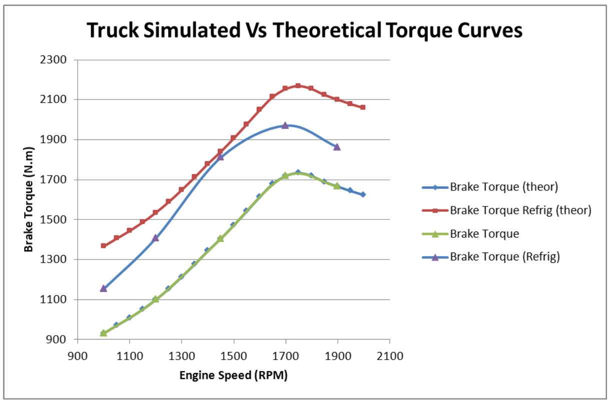

Figure 18. Sample truck engine simulated vs. theoretical torque curves

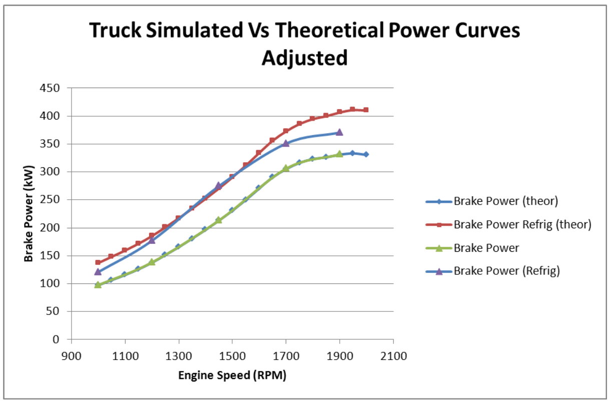

Figure 17. Sample truck engine adjusted simulated vs. theoretical power curves

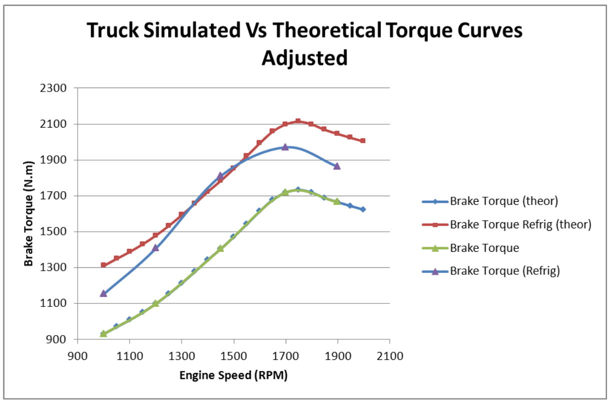

Figure 19. Sample truck engine adjusted simulated vs. theoretical torque curve

In terms of the increase in brake torque output, the theoretical increase was calculated to be 26.11% and the simulated increase was found to be 21.39%. The difference in these two figures was believed again to be due to losses which were not taken into account by the Otto cycle calculations and the assumption that the combustion temperature would remain the same, as when this temperature was adjusted the calculated performance gain dropped to 22.76%. The deviation between the theoretical and simulated power curves was found to be 0.02% for the torque curve of the standard engine, 9.07% for the torque curve of the engine when the refrigerated intercooler was installed and 6.67% for the same power curve when the reduction in combustion temperature was taken into account. The theoretical and simulated torque curves for the sample truck engine are shown in Figure 18. The theoretical and simulated torque curves with the adjusted theoretical refrigerated torque curve are shown in Figure 19.

7. Reduced fuel results

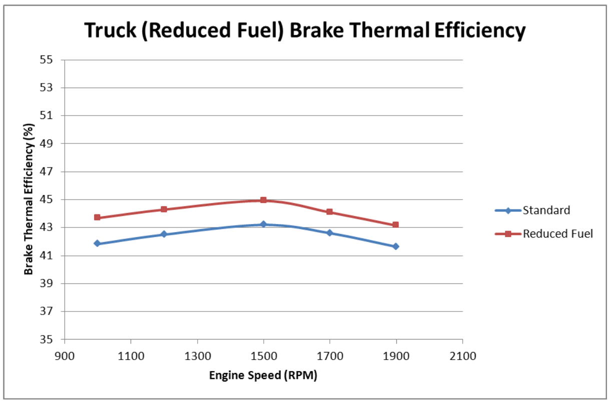

Figure 20. Sample truck engine (reduced fuel) simulated brake thermal efficiency

When the power output of the truck engine with the refrigerated intercooler installed was reduced to the same level as the power output of the standard engine by reducing the air/fuel ratio, certain improvements in efficiency were observed. The brake thermal efficiency increased by 3.95% and the required fuel volume flow reduced by 1.9%. These were small but worthwhile improvements on the efficiency of the standard engine. The results for the fuel volume flow and brake thermal efficiency for both engines are graphed against engine speed in Figures 20 and 21.

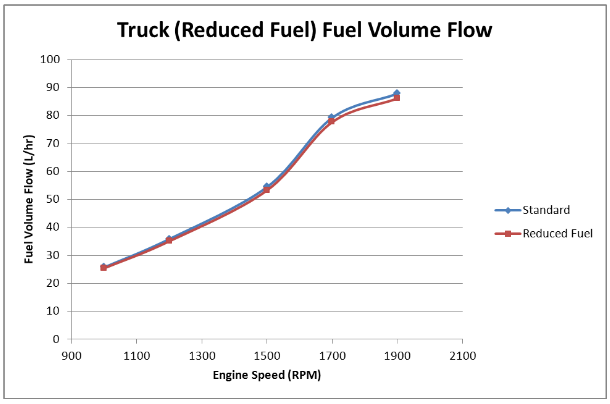

Figure 21. Sample truck engine (reduced fuel) simulated fuel volume flow

8. Actros engine simulated performance

As was mentioned in the Calculations section, the simulation of the Actros engine was unsuccessful, as an accurate model of the engine could not be created in the time available to do so. However, the success of the comparison of the simulated and theoretical results between the other models would suggest that the potential performance improvements could be not too far off what was predicted by the theoretical calculations.

9. Conclusion

The theoretical analysis and simulation results indicated that the proposed system of refrigerated intercooling could be used to significantly increase the power and torque output of an engine and, to a lesser degree, to improve the engine's efficiency. Therefore it can be said that the main scope for further research and testing in this area would be for applications where power and torque benefits are important and efficiency benefits are not of paramount importance. Nonetheless, this system could yield small efficiency benefits and perhaps, with further research, these could be improved. Therefore, this project could be seen as having opened up two potential lines of research.

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