TRABAJO DE FIN DE MASTER

Investigación específica sobre sistemas de EGR y Sobrealimentación en MCIA

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1. INTRODUCCIÓN TRABAJO FINAL DE MASTER

1.1 Introducción.

El presente Trabajo Final de Máster (TFM) muestra la experiencia adquirida en el grupo CMT – Motores Térmicos en el ámbito de los motores diesel de inyección directa sobrealimentados desde el punto de vista de **Renovación de la Carga**.

En la evolución actual de los motores de combustión, los fabricantes e investigadores se enfrentan a dos retos impuestos por la sociedad y lo usuarios de vehículos. Dichos retos son por una parte la reducción de emisiones contaminantes, reguladas por las cada vez más restrictivas normativas gubernamentales y por otro lado la mejora de las prestaciones del motor, evaluadas por los usuarios en los parámetros de *potencia* y *consumo*. De forma general, y como se verá en a lo largo de este TFM, se podría decir que ambos retos son opuestos y que la mejora en uno de ellos afecta negativamente en el otro.

La gran aceptación de los motores Diesel en el mercado por los buenos resultados en cuanto a prestaciones del motor, es producto de los avances tecnológicos introducidos en cuanto a sistemas de inyección directa de alta presión y las posibilidades de elevada **sobrealimentación,** mejorando la Presión Media Efectiva de los motores a la vez que se reduce su consumo específico.

Sin embargo, una de las grandes limitaciones de los motores Diesel actuales son las emisiones de NOx. Para reducirlos se utiliza la técnica de la **recirculación de los gases de escape** (**EGR**), que si bien es efectiva para la reducción de dichas emisiones, afecta directamente a las prestaciones del motor, por su influencia en la combustión y el proceso de Renovación de la Carga.

Ante las futuras normativas de emisiones y la demanda ciudadana de motores con menores consumos, sin perder capacidad de reacción, los fabricantes e investigadores de motores Diesel están trabajando por mejorar los niveles de sobrealimentación y la efectividad del EGR.

De esta forma, en este TFM se recogen algunos trabajos desarrollados sobre estos dos aspectos dentro del proceso de renovación de la carga de un motor. Por un lado se abordan las diferentes técnicas para realizar recirculación de gases de escape (EGR) y por otro se realiza un estudio sobre el comportamiento compresor con una restricción a su entrada para la sobrealimentación de un motor diesel.

1.2 Objetivo, Metodología y Medios.

Objetivo TFM:

Profundizar en los efectos de las nuevas técnicas de recirculación de gases de escape y de las condiciones de trabajo específicas del compresor sobre el proceso de Renovación de la Carga de motores Diesel sobrealimentados.

Metodología TFM:

La metodología de trabajo tiene como antecedente los trabajos del CMT - Motores Térmicos desarrollados en el ámbito de renovación de la carga y de forma concreta en los proyectos y tesis doctorales sobre EGR. Además de un proceso de revisión del estado del arte en que se encuentra la comunidad científica respecto a estos temas.

Para desarrollar estas investigaciones, se han de llevar a cabo dos metodologías complementarias. Por un lado modelización del motor en un modelo de acción de ondas. Por otro lado, el ensayo experimental de los sistemas investigados.

Medios TFM:

Los medios empleados para la consecución de estas investigaciones han sido los facilitados por CMT - Motores Térmicos.

El modelado de los motores se ha realizado con WAM (Wave Action Model) un programa de modelado unidimensional desarrollado expresamente en CMT - Motores Térmicos para el estudio de las líneas de admisión y escape.

Los ensayos experimentales se han realizados en las salas de ensayo de CMT - Motores Térmicos. Dichas salas de ensayo están equipadas con los elementos de medida necesarios para la caracterización de los componentes ensayados.

2. ESTUDIOS DESARROLLADOS PARA EL TRABAJO FINAL DE MASTER.

2.2 Implementación y análisis de la combinación de sistemas de EGR de alta y baja presión

- Objetivos:

1- Analizar los efectos de los sistemas de EGR de alta presión utilizado actualmente en motores Diesel de inyección directa sobrealimentados y de EGR de baja presión sobre los procesos de renovación de la carga y su eficacia respecto a las emisiones contaminantes.

2- Comprobar la posibilidad de realizar la combinación de dos sistemas EGR, de alta y de baja presión. Analizar el proceso de renovación de la carga y estudiar el efecto sobre el turbogrupo, las emisiones contaminantes del motor y las prestaciones.

- Metodología:

En esta investigación se llevan en paralelo dos líneas de trabajo, el modelado y el ensayo experimental. Ambos trabajos ofrecen información de forma individual y a la vez complementaria.

Los trabajos de modelado se han desarrollado con el programa de acción de ondas (WAM, Wave Action Model) propio de CMT-Motores Térmicos. Dicho programa permite realizar el modelado unidimensional de los sistemas de admisión y de escape. El cual permite, una vez ajustados los parámetros termofluidodinámico a los del motor estudiado, poder analizar variables difíciles de medir en el motor real (gastos y temperaturas instantáneas, trabajo de bombeo de motor, etc.) y realizar estudios paramétricos con los distintos componentes de los sistemas de admisión y escape.

Los ensayos se han realizado en las salas de CMT - Motores Térmicos dispuestas a tal efecto. En ellas el motor es conectado a un freno eléctrico que permite regular el régimen de giro y el par. Mediante el acceso a la configuración de la ECU se modifica la gestión electrónica del motor, para regular el comportamiento de los distintos componentes del motor (turbina, EGR, temperaturas de funcionamiento, etc.). Finalmente, diferentes condiciones de funcionamiento del motor (régimen de giro, par motor y grado de carga) se ensayan para ver la evolución y el comportamiento de los distintos elementos del sistema de renovación de la carga. En los trabajos sobre EGR las condiciones de funcionamiento ensayadas corresponden a puntos de funcionamiento del motor de un ciclo de homologación según la normativa europea de emisiones contaminantes.

- Resumen:

La recirculación de los gases de escape (EGR, Exhaust Gas Recirculation) es una técnica utilizada en los motores de combustión interna alternativos para reducir las emisiones de NOx. Dicha técnica consiste, básicamente, en la extracción de gases quemados de la línea de escape del motor e introducirlos en la línea de admisión.

El EGR actúa como un gas inerte en el interior del cilindro, inhibiendo las reacciones de formación del NOx que tienen lugar durante el proceso de combustión. Al actuar directamente sobre el origen de las emisiones de NOx hace que el EGR sea una práctica muy efectiva para reducirlos. Sin embargo, también afecta al proceso de combustión de forma desfavorable para otras emisiones, como la emisión de partículas, y sobre las prestaciones del motor aumentando el consumo específico.

Por otra parte, para conseguir el gasto de EGR necesario se han ido desarrollando diferentes arquitecturas del sistema de EGR, que tendrá una influencia directa sobre el proceso de renovación de la carga y afectaran directamente a los componentes de admisión y de escape.

De entre las muchas arquitecturas que se pueden implementar para realizar el EGR, es el EGR de Alta Presión refrigerado el que actualmente se utiliza en todos los vehículos. Este tipo de EGR consiste en la extracción de gas de escape antes de entrar en la turbina, refrigerarlo con agua de motor en un intercambiador agua-gas y mezclarlo con el aire fresco en el colector de admisión después del compresor. Sin embargo este sistema tiene limitaciones, sobre todo, las que le impiden alcanzar altas tasas de EGR para lograr alcanzar los niveles de emisiones contaminantes de las normativas futuras.

El endurecimiento de las normativas anticontaminación está obligando a que la gestión del EGR sea más precisa. No solo con vistas a reducir más las emisiones de NOx, que es su misión principal, sino también para evitar perjudicar las emisiones de PM y las prestaciones del motor.

Diversos estudios sobre diferentes alternativas al EGR de Alta Presión para alcanzar dichos niveles de emisiones han surgiendo en los últimos años, entre las que se encuentran las técnicas de post-tratamiento de NOx o el EGR de Baja Presión. Si bien los post-tratamientos son efectivos en la reducción de NOx no es una tecnología muy desarrollada todavía, su entrada en motores comerciales parece cercana.

Por otro lado aparece el EGR de Baja Presión, una técnica que hace unos años fue descartada por cuestiones de durabilidad de componentes ya que el gas de escape (sucio de partículas) tenía que atravesar el compresor y el intercambiador del motor. La extensiva introducción de los filtros de partículas en la línea de escape de los motores diesel sobrealimentados, hace que el gas de escape ya no sea un peligro para estos componentes y el EGR de Baja Presión vuelve a ser una alternativa. Con él se logran alcanzar altas tasas de EGR y mejoran las emisiones de PM y el comportamiento motor. Sin embargo, la condensación del vapor del agua del EGR debido a la refrigeración y la necesidad de instalar una restricción en la admisión o en el escape que garantice el salto de presión necesario para que exista el gasto de EGR deseado, son las principales limitaciones de esta arquitectura.

Como se ha comentado anteriormente la aplicación de sistemas de post-tratamiento de NOx es efectivo, pero parten de aplicar cierta tasa de EGR. Por otra parte, los motores con nuevos modos de combustión requieren de altas tasas de EGR y alta capacidad de control para su desarrollo. Si además se tiene en cuenta que hoy por hoy el EGR es el único medio para reducir NOx, se justifica la continuidad de los estudios de EGR. El estudio que aquí se presenta, está encaminado a mejorar las emisiones de NOx y reducir la influencia en las prestaciones del motor mediante la combinación de los sistemas de EGR de alta y baja presión, de forma que se aprovechen sus ventajas individuales.

La trayectoria de las investigaciones sobre EGR llevadas a cabo en CMT-Motores Térmicos, es paralela a la evolución del EGR en motores diesel sobrealimentados. Desde estudios iniciales sobre EGR para vehículos industriales, pasado por sistemas de EGR interno donde se fuerza el reflujo entre la admisión y escape mediante el cruce de válvulas, la optimización y gestión de los actuales sistemas de EGR de Alta Presión o el estudio de las nuevas tendencias de EGR de Baja Presión y EGR altamente refrigerado. Todos ellos respaldados por el interés no solo de comunidad científica sino también de los fabricantes de motores que solicitan las investigaciones.

En el artículo 1 se detalla todo el proceso, el planteamiento y los resultados de la investigación en la que se han combinado los sistemas de EGR de Alta Presión y Baja Presión, donde el EGR de Baja Presión no utiliza un sistema de restricción para garantizar el gasto de EGR entre escape y admisión. Se han realizado ensayos de dos puntos de funcionamiento del motor, correspondientes con puntos de funcionamiento del ciclo europeo de homologación, y para cada uno de ellos un barrido con la combinación de ambos sistemas. Después se han evaluado los resultados obtenidos respecto a los parámetros de renovación de la carga y tasa de EGR. Además también se estudian los resultados de emisiones de NOx y Partículas y las prestaciones de motor, evaluadas por el consumo específico.

Entre las conclusiones obtenidas de este artículo cabe destacar dos de ellas, de forma muy resumida y general. Por un lado la de que es posible utilizar ambos sistemas de forma conjunta y que las mayores tasas de EGR se dan con ambos sistemas trabajando conjuntamente, como se ve en la figura. Por otro lado, se obtiene una zona de funcionamiento conjunto, descrita en la figura, en la que se optimizan las emisiones de NOx y PM.

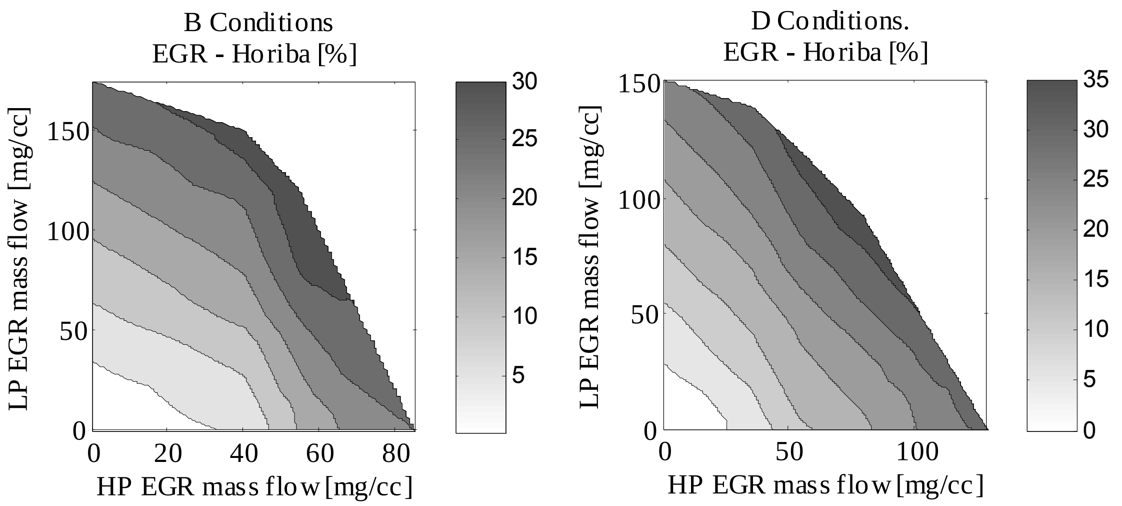

Figura 1. Tasa de EGR para los puntos de funcionamiento B y D con las distintas combinaciones de EGR HP y EGR LP

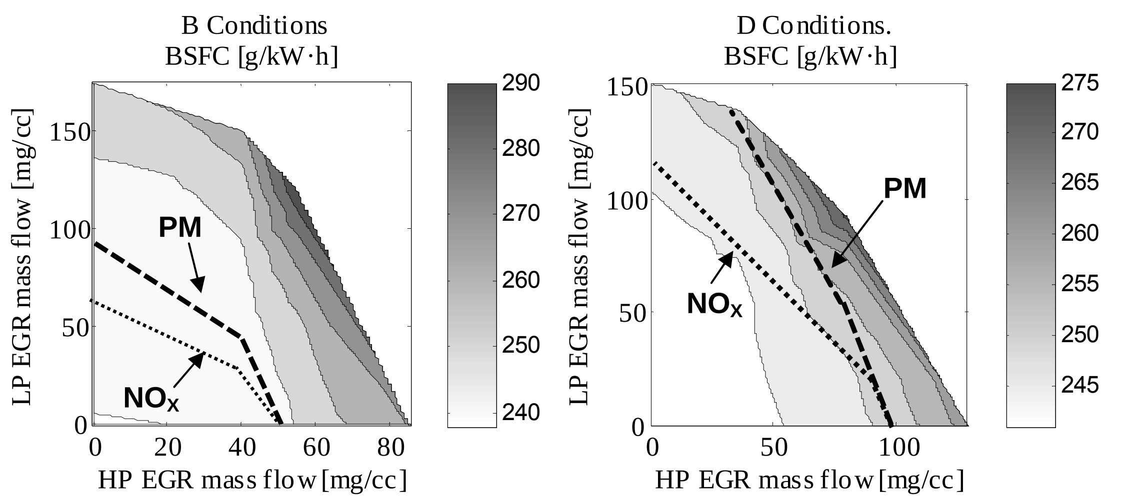

Figura 2. Gasto Específico para los puntos B y D con las distintas combinaciones de EGR HP y EGR LP. Limites de máximas emisiones de PM (trazo discontinuo) y NOx (trazo de puntos), las zonas limitadas por ambas líneas son las combinaciones que optimizan las emisiones de PM y NOx.

- Conclusiones:

1- El sistema de EGR de Alta Presión utilizado en actualidad tiene limitada su capacidad de alcanzar grandes tasas de EGR, entre otros motivos, debido a la alta temperatura de admisión. Además, las altas tasas de EGR alcanzadas con el sistema de alta presión tienen un efecto indeseado sobre las emisiones de PM y sobre las prestaciones del motor.

2- Los sistemas de EGR de Baja Presión logran mayores tasas de EGR que los de Alta Presión. Además los efectos sobre las emisiones de PM y sobre las prestaciones del motor aparecen con tasas mayores. Sin embargo, las altas tasas de EGR de Baja Presión son alcanzadas introduciendo una restricción en la línea de escape o de admisión que garantice el salto de presión suficiente entre ambas líneas para que haya el gasto necesario de EGR. Además, en algunos puntos de funcionamiento del motor, los altos niveles de refrigeración de EGR de Baja Presión dan lugar a condensaciones del vapor de agua generalmente de carácter ácido.

3- La combinación de ambos sistemas permite alcanzar mayores tasas de EGR que de forma individual, si no usamos restricción para realizar el EGR de baja presión. Además se puede definir una zona de trabajo de forma que se puedan utilizar las ventajas de ambos sistemas en circunstancias específicas. Garantizando los niveles de emisiones de NOx y PM, así como las prestaciones del motor. (Artículo 1)

2.3 Estudio del comportamiento de un motor diesel de inyección directa sobrealimentado cuyo turbocompresor trabaja con una tobera de restricción.

- Objetivos:

1- Estudiar el comportamiento de un motor diesel sobrealimentado de inyección directa y 2.0 litros de cilindrada, cuyo turbocompresor lleva una tobera de restricción en su entrada.

2- Obtener el mapa del compresor trabajando con dicha restricción.

- Metodología:

Como se plantea en la metodología general y en el caso anterior, en este trabajo también se llevan a cabo en paralelo los trabajos de modelado y de ensayo experimental.

Mediante el modelado se realiza un análisis del comportamiento fluidodinámico de las líneas de admisión y escape. Una vez ajustado al comportamiento del motor real, permitirá realizar ensayos paramétricos modificando algunos componentes de la línea de admisión y ver como varía el proceso de renovación de la carga.

Desde el punto de vista de los ensayos experimentales, se han realizado dos tipos de ensayos. En primer lugar están los ensayos de motor para validar el modelo realizado y analizar el comportamiento del motor. En estos ensayos se miden presiones tanto instantáneas como medias de admisión, escape y la presión instantánea en la cámara de combustión..... Por otro lado está el ensayo del compresor con una tobera de restricción instalada en la entrada para la obtención de su mapa de funcionamiento, para a continuación compararlo con el mapa sin usar restricción.

- Resumen:

Hasta hace algunos años (principios de los noventa) los motores Diesel de pequeña cilindrada, motores de encendido por compresión (MEC), no eran una alternativa a los motores de encendido provocado (MEP) en cuanto a prestaciones de alta potencia específica y respuesta en transitorio. Los MEC eran valorados sobre todo por su bajo consumo.

La potencia efectiva del motor (para una cilindrada dada) podemos obtenerla a partir de régimen de giro o de pme. Sin embargo los MEC tienen en el régimen de giro una de las limitaciones más importantes debido a la lentitud relativa de los procesos de mezcla y combustión.

Con los últimos avances introducidos en los MEC, se han mejorado las prestaciones del motor manteniendo su propiedad de bajo consumo. De forma que han pasado a ser competitivos en el mercado de vehículos, incluso superando en ventas a los motores MEP.

Esta mejora de los MEC es debida fundamentalmente a los altos niveles de sobrealimentación que se están desarrollando y los nuevos sistemas de inyección directa de alta presión, que mejoran los procesos de mezcla y combustión y permiten obtener una elevada pme del motor.

Esta evolución de los motores MEC ha hecho que no solo sean interesantes para usuarios de la calle sino también en el ámbito de la competición. Sin embargo, dado que la sobrealimentación puede marcar diferencias muy grandes entre los competidores, los reglamentos de la competición limitan las prestaciones de los motores obligando a instalar a la entrada del compresor una tobera de restricción. El estudio aquí presentado trata sobre motor diesel de inyección directa de 2.0 litros de cilindrada sobrealimentado. Donde, a la entrada del compresor se ha colocado una tobera de restricción de 35 mm de diámetro. El estudio se desarrolla en dos vertientes:

A. Estudiar mediante modelado unidimensional en condiciones estacionarias del motor, el proceso de renovación de la carga para analizar el comportamiento de la tobera.

B. Obtener el mapa del compresor trabajado con una restricción a la entrada. Artículo 2.

A. Modelado fluidodinámico del motor

Para el modelado del motor se ha utilizado el modelo de acción de ondas WAM. El modelo ha sido ajustado con los ensayos efectuados en banco del motor en puntos de funcionamiento estacionarios registrándose para cada ensayo las características del fluido en los colectores de admisión y escape del motor.

De estos ensayos ha sido posible definir unos parámetros para la ley de combustión, de forma que, tanto la presión en cámara de combustión como la energía del gas de escape han sido adecuadamente modeladas.

Comparando los resultados de este modelo con los resultados medidos comprobaremos la calidad del ajuste. En la tabla se comparan los resultados medios para el caso de 3500 rpm. En la gráfica se muestran los resultados instantáneos de presión en la línea de admisión, en la línea de escape y la presión en la cámara de combustión.

Tabla 1. Comparativa de resultados medios, medidos y modelados

	RPM	P admisión mmbar	P escape mmbar	P tobera mmbar	RPMTurbo $x10^3$	T admisión ºC	T escape ºC
Ensayo	3502.5	3802	4124	541	196.9	48.5	997
Modelo	3500	3822	4242.4	640	200.46	38.87	989.64

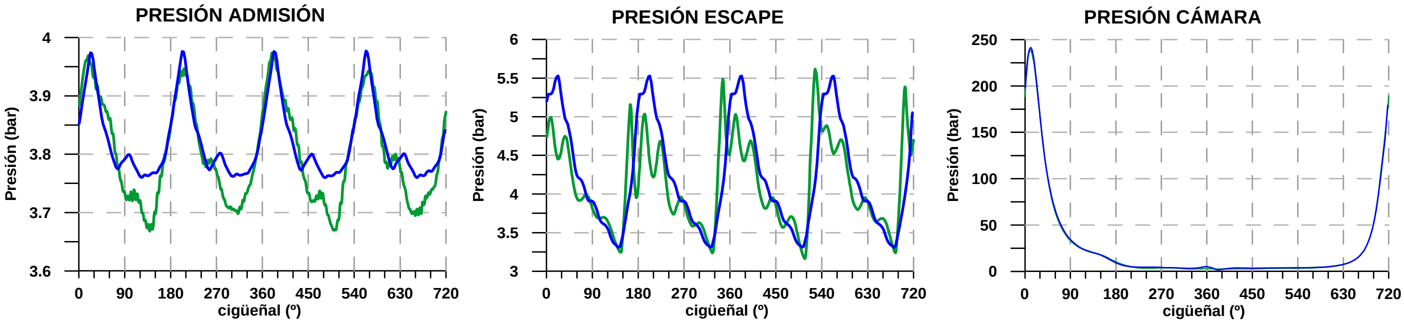

Figura 3. Comparativa de resultados instantáneos. Medidos (azul) y modelados (verde)

Una vez obtenido el modelo se puede visualizar la evolución instantánea de aquellos parámetros que son difíciles de medir en una instalación, como se ve en la figura 4. En este caso se representan los gastos instantáneos de aire o gas de escape a través de los componentes de admisión y escape y además el régimen de giro del turbocompresor.

Los gastos por las válvulas nos muestran cómo se desarrolla el proceso de renovación de la carga en el cilindro. El gasto por la garganta es el gasto que está trasegando el motor. El dato más característico es que el gasto por la garganta no trabaja siempre en condiciones sónicas, es pulsante, y por tanto no se le está aportando al motor el máximo gasto de aire posible.

Los trabajos de potenciación del motor, desde el punto de vista de renovación de la carga, deben ir orientados a logra que el motor reciba la máxima cantidad de aire posible, lo que significa que la tobera trabaje siempre en condiciones sónicas.

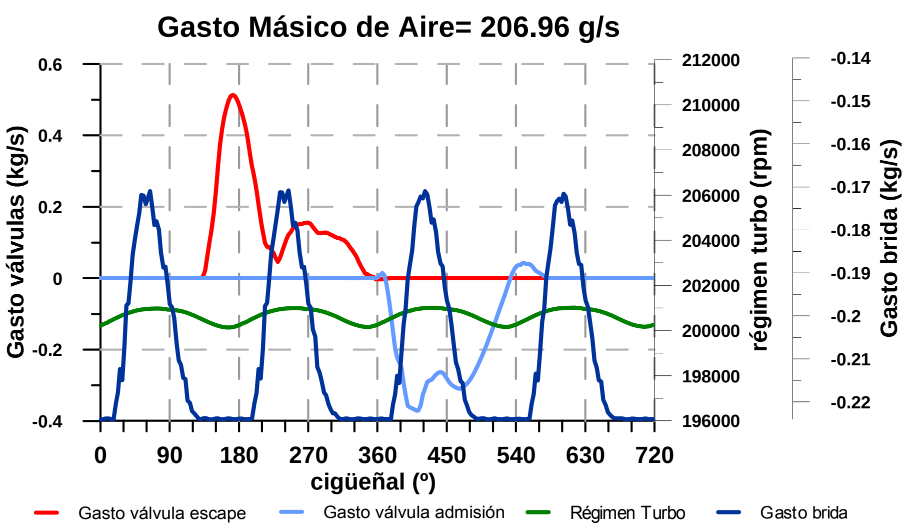

Figura 4. Evolución instantánea de los gastos de fluido por las válvulas, el gasto en la garganta de la tobera y el régimen del turbocompresor.

B. Obtención del mapa de funcionamiento de un compresor con una tobera de restricción en su entrada.

Los resultados de este estudio fueron utilizados para la redacción del Artículo 2 presentado en este TFM. En el artículo se desarrolla detalladamente todo el planteamiento del estudio, la metodología de trabajo seguida, los procedimientos de ensayo, la descripción de las salas de ensayo, el mapa compresor obtenido y las conclusiones finales de dicho estudio.

La conclusión más importante que se obtiene y que afecta al comportamiento del motor estudiado, es que la restricción en la entrada del compresor desplaza la línea de bombeo del compresor hacia la izquierda del mapa inicial, manteniendo el rendimiento del compresor. Esto implica que para los gastos bajos de aire que trasega el compresor se pueden alcanzar mayores relaciones de compresión.

En la figura 5 se representan de forma comparativa los mapas del compresor trabajando con la tobera (en negro) y del compreso original sin la tobera (en gris), para mostrar dicho efecto.

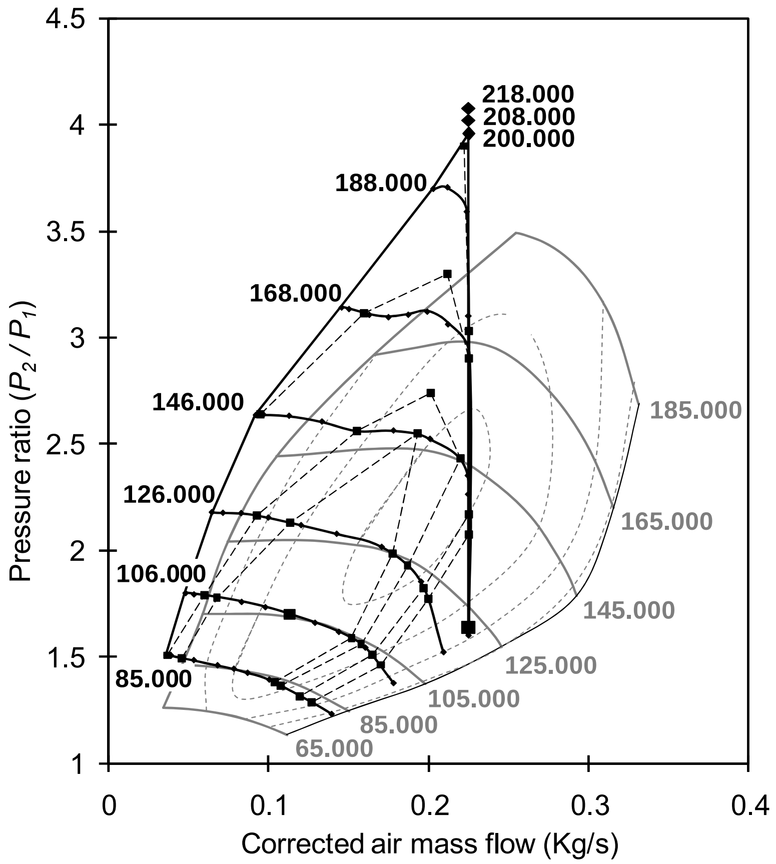

Figura 5. Comparativa de los mapas del compresor trabajando con la tobera de restricción (negro) y sin ella (gris).

- Conclusiones:

1- El gasto medio trasegado por el motor podría ser aumentado eliminando el flujo pulsante en la garganta de la tobera, de forma que siempre trabajara en condiciones sónicas.
2- El uso de una tobera de restricción, con geometría convergente-divergente, en la entrada de un compresor mejora el comportamiento del turbocompresor en la zona de bombeo. Aunque tobera limita la capacidad máxima de trabajo del compresor, la línea de bombeo se desplaza hacia la izquierda del mapa, lo que significa que para gastos de aire bajos. (Artículo 2)

On the combination of High Pressure and Low Pressure EGR loops for improved fuel economy and reduced emissions in HSDI engines

Abstract

The reduction of NO_x emissions is one of the most challenging problems in current Diesel engines. In this paper, an experimental study of the combination of Low Pressure and High Pressure EGR architectures has been carried out. The effects of both EGR architectures and their combination on engine performance and emissions have been analysed. The results have shown that the Low Pressure configuration improves High Pressure EGR results in BSFC, NO_x and exhaust gas opacity, nevertheless the HC emissions are increased, especially during the engine warm up. In addition, the EGR rate that can be achieved with Low Pressure EGR systems is limited by the low pressure difference between DPF outlet and compressor inlet, and the High Pressure system can be used to increase the EGR rate to the required levels without increasing pumping losses. In this sense, the combination of both EGR layouts offers significant advantages to reduce emissions and fuel consumption to meet future emission requirements

1. Introduction

The increase in the specific power of DI Diesel engines during the last 20 years and their higher efficiency has allow them to reach a favourable position in the market of light duty vehicles. In addition, as far as the Global Warming problem is concerned, their lower CO_2 emissions are an additional advantage for DI Diesel engines. Nevertheless, the environmental concern has also pushed governments to progressively reduce the vehicle emission limits. Despite the stricter emissions laws, road transport was responsible in 2006 for 17,7 % of all EU-27 greenhouse gases (GHG) emissions, apart from providing 40 % of NO_x emissions, 36 % of CO emissions and 18 % of the emission of non-methane volatile organic compounds [1]. Then, important efforts should still be done in order to strongly reduce vehicle emissions. In this sense, the development of powertrains with low environmental impact is one of the most challenging problems to be faced during the next decades [2].

In the case of Diesel engines, particles and NO_x have been traditionally the most challenging pollutants to control. Regarding particles, while the air-to-fuel ratio control and the optimization of injection strategies were traditionally sufficient to avoid excessive particulate emissions, during the last years the use of DPF has been a widespread solution to fulfil emissions regulations. On the other hand, despite the development in NO_x after-treatment for Diesel engines, EGR remains a cost-effective solution to fulfil current and future emissions regulations. Also, during the last decade several combustion modes such as HCCI (Homogeneous Charge Compression Ignition), PCCI (Premixed Controlled Compression Ignition) or MKI (Moduled Kinetics) have been developed in order to strongly reduce emissions [3, 4]. It should be noted that the EGR is a key technique to control such combustion processes [5]. Therefore, according to the previous aspects it is expected that the EGR will continue playing a major role in DI Diesel engines.

Regarding the EGR, despite different methods to reintroduce part of the exhaust gases in the engine cylinders have been developed, the High Pressure EGR approach is by far the most commonly employed EGR architecture in current engines [6]. It is known as external EGR because the burnt gas is extracted from the exhaust to be introduced in the intake line. On the contrary, internal EGR consists on avoiding the exhaust of part of the burnt gas that stays into the cylinder to participate in the next combustion process. External EGR systems have prevailed due to their better flow and temperature regulation. The terms “High Pressure” refers to the gas is extraction from the exhaust manifold (upstream the turbine) and introduction in the intake manifold (downstream the compressor), in this sense the EGR line is at a higher pressure than ambient. According to the system layout, the EGR rate is limited by the pressure difference between the intake and exhaust manifolds. In addition, since the turbocharger behaviour also depends on the intake and exhaust conditions, a strong coupling between both systems appears. Other problems attached to the HPEGR systems are the difficulty to provide a homogeneous intake charge between cylinders [7] or the important increment in intake temperature despite employing EGR coolers [8].

The Low Pressure EGR (LPEGR) technique is a topical subject because some of the problems of the HPEGR systems can be reduced by modifying the EGR layout [9, 10, 11]. The LPEGR method consists on extracting part of the exhaust gas from the DPF outlet and guiding it to the compressor inlet. At these conditions, the recirculated gas has not solid particles and its temperature is lower than that obtained at the turbine inlet. In addition, the coupling between turbocharging and EGR systems is reduced due to the fact that the whole exhaust gas flows through the turbine, and then the increase in EGR rate does not involve a reduction in the turbine available energy. The EGR rate with the LPEGR system does not depend on the intake and exhaust pressures, nevertheless, despite the fact that the compressor inlet pressure is always lower than the DPF outlet pressure, the pressure drop between those points is usually not high enough to reach the required EGR levels. In this sense, LPEGR systems require a backpressure valve at the DPF outlet (downstream the EGR extraction) or an intake throttle at the compressor inlet (upstream the EGR injection). Since the EGR is introduced at the compressor inlet, there is enough length in the intake line to achieve a perfect air and EGR mixture before the cylinders.

Of course, the LPEGR system involves some difficulties. Since the EGR goes through the whole intake line the use of DPFs is strictly necessary in order to prevent the exhaust particles from damaging the compressor wheel and also plugging coolers. With the widespread application of DPFs, the compressor and charge air cooler reliability problems with LPEGR, have been reduced. Other problems of the LPEGR system are the condensation of species contained in the burnt gas along the intake line, and especially at the charge cooler.

According to the weak and strong points of both systems, the objective of this work is to evaluate the benefits introduced by the combination of both EGR layouts, taking into account three main aspects:

1. The capability of both systems to introduce mass into the cylinder assessed by the air mass flow and the EGR rate.
2. The potential of each system for the pollutant emissions reduction, taking into account the effects of the EGR on both the composition and temperature of the intake charge.
3. The effects on engine performance, especially on BSFC, due to the differences on intake charge composition and temperature, but also the effect on pumping losses of both LP and HPEGR approaches.

2. Experimental Set Up and Methodology

In the present paper, the study of the effects of the LP and HPEGR combination on engine performance and emissions has been approached experimentally. The study has been performed on a state-of-art 2.0 litre HSDI Diesel engine. The engine, which main characteristics appear in table 1, was equipped with VGT, intercooler, DPF and a cooled High Pressure EGR loop.

table 1. Engine features

Architecture	4 L
Displacement	1998 cm3
Bore x Stroke	85 x 88 mm
Valves	4 / cylinder
Compression ratio	18:1
Turbocharger	Garret VNT GT 1749V
EGR	High Pressure Cooled
After-treatment	Oxy-catalyst + DPF
Max. Power / speed	100 kW - 4000 rpm
Max. Torque / speed	320 Nm - 1750 rpm

As shown in figure 1 a Low Pressure EGR circuit was added to the original engine. An open code ECU was used to modify the engine calibration. Also, the engine was fully instrumented to measure temperatures and pressures in different interesting engine locations.

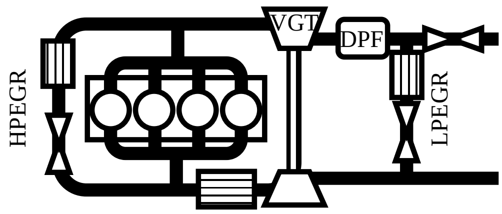

figure 1. Experimental set up.

Two engine operating points were defined for testing, they were selected from the conditions reached during the NEDC cycle in order to be representative of real operation. In this sense, the engine conditions achieved when a given vehicle reaches 100 and 120km/h during the MVEG-A cycle were tested due to its important contribution to the NO_x and PM emitted over the entire NEDC cycle. A summary of those conditions can be found in

.

table 2. Tested conditions

	engine speed (rpm)	fuel (mg/cc)	WCAC outlet T (ºC)	intake p (bar)	base line conditions (HPEGR)			
					EGR (%)	M (Nm)	NOx (ppm)	Opacity (%)
A	1870	16.3	35	1.066	23.6	72	123	
B	2250	22.8	35	1.190	11.3	100	240	

For both operating conditions, tests consisted on performing a sweep of EGR rates combining the use of both LP and HP EGR systems, keeping the values of engine speed, injected fuel, WCAC outlet temperature and intake pressure constant.

For each engine point, the EGR ranged from 0% (no EGR) to the maximum EGR rate the engine can admit. The maximum EGR rate limits were established by two different phenomena:

1. Unsteady combustion: An extremely high EGR rate leads to an unacceptable increase in BSFC due to the high delay in the combustion process.

2. Recirculated mass flow limited by the pressure ration between the source and discharging points: In the case of the LPEGR system, the low pressure difference between DPF outlet and compressor inlet introduces an important limitation in the maximal EGR rate that can be achieved without auxiliary devices such as valves at the DPF outlet to increase the gas pressure at the EGR inlet.

2.1 HP and LPEGR flow estimation

The estimation of HP and LPEGR flows starts with the mass balance in the intake manifold:

$$\dot{m}_{air} + \dot{m}_{LP} + \dot{m}_{HP} - \dot{m}_{engine} = \frac{\partial m_{manifold}}{\partial t} \tag{1}$$

Assuming that the intake charge behaves as perfect gas, the previous equation can be written as:

$$\dot{m}_{air} + \dot{m}_{LP} + \dot{m}_{HP} - \dot{m}_{engine} = \frac{V}{RT}\frac{\partial p}{\partial t} - \frac{pV}{RT^2}\frac{\partial T}{\partial t} \tag{2}$$

Since the present study only considers steady state behaviour, the variations of both intake manifold pressure and temperature can be neglected, arriving to:

$$\dot{m}_{air} + \dot{m}_{LP} + \dot{m}_{HP} - \dot{m}_{engine} = 0 \tag{3}$$

The mass flow aspirated by the engine is calculated taking into account that the engine volumetric efficiency does not depend on the EGR rate. In this sense, the mass flow admitted by the engine at a given speed and load conditions is considered to be constant and is obtained from the test without EGR. In addition, a sensor located upstream the LPEGR introduction provides the air mass flow signal. The mass flow through any of the EGR valves can be modelled by the nozzle flow equation:

$$\dot{m}_{egr} = \frac{A(\chi_{EGR})p_u}{\sqrt{RT_u}}\pi^{\frac{1}{\gamma}}\sqrt{\frac{2\gamma}{\gamma-1}\left(1-\pi^{\frac{\gamma-1}{\gamma}}\right)} \tag{4}$$

where χ_{EGR} is the EGR valve position, T_u and p_u are the temperature and the pressure upstream the valve, A is the valve effective area, which depends on the valve position, and π is the pressure ratio in the valve. Despite the production engines do not have the sensors required to calculate the EGR flows from equation (4), in the experimental facility used in this study they are available. Nevertheless, it should be noted that equation (4) provides a poor estimation of the LPEGR flow due to the very low pressure drop across the LPEGR system. In this sense, equation (4) has been used to calculate the flow through the HPEGR loop, while the LPEGR flow has been obtained from equation (3). In figure 2, the EGR rate measured with the gas analyser is plotted versus the proposed estimation. The estimated EGR rate is defined as:

$$EGR = \frac{\dot{m}_{LP} + \dot{m}_{HP}}{\dot{m}_{LP} + \dot{m}_{HP} + \dot{m}_{air}} \tag{5}$$

As can be observed in figure 2, there is a well correlation between the measured and the estimated EGR rate, nevertheless, it can be stated that there is a light overprediction of the proposed method.

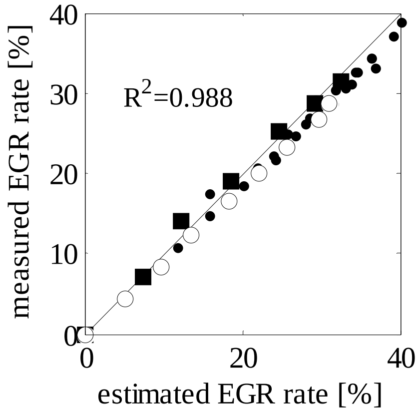

figure 2. Measured EGR rate versus estimated EGR rate at A conditions. ○: LPEGR. ■: HPEGR. · :HP and LPEGR.

3. Results

3.1 Considerations on the effects of HP and LPEGR

One or the main differences between HP and LPEGR systems lies in the conditions of the intake charge. When the LPEGR is used, the intake temperature is almost independent of the EGR rate, nevertheless, with the HPEGR an increase in the EGR rate leads to an inevitable increase in the intake temperature. The left part of figure 3 shows how the increase in the HPEGR rate from 0 to 30% produces an increase in the intake temperature of almost 80ºC. Since tests have been performed with constant intake pressure, the increase in temperature involves a 20% reduction in the intake density. Then, in addition to the substitution of part of the air charge by burnt gasses, the HPEGR produces an additional reduction in the air charge due to the density reduction (“thermal throttling” [12]). In the right part of figure 3 the reduction in the intake O_2 concentration with the EGR can be observed. For a given EGR rate, the HPEGR system produces an additional reduction in the O_2 concentration due to the increase in the intake temperature [12].

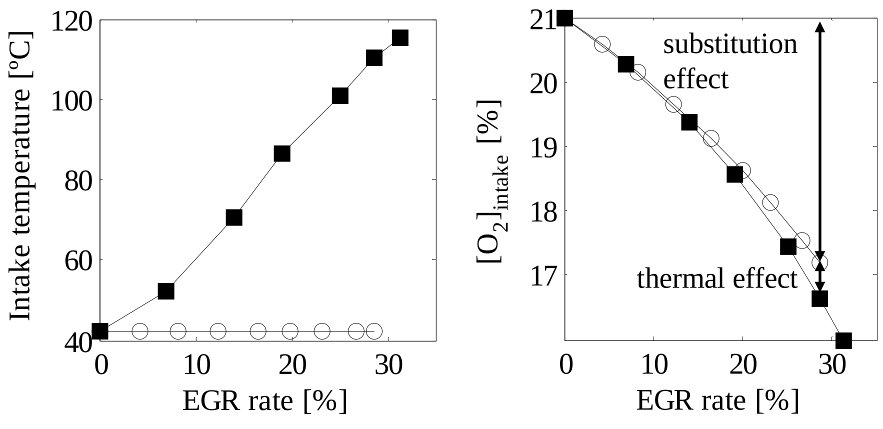

figure 3. Effect of the EGR rate on the intake temperature (left) and intake O2 concentration (right) at A conditions.
○: LPEGR. ■: HPEGR.

As far as the EGR has effects on the intake charge properties, it affects the combustion process. The reduction in the intake O_2 concentration contributes to the increase in the ignition delay. In this sense, figure 4 shows a shift in the rate of heat release pattern in the cases with EGR. The higher ignition delay with EGR provides more time for the fuel to mix with the oxidizer, which increases the amount of pre-mixed fuel. Comparing the cases with EGR, it can be observed a higher rate of heat release with the LPEGR system. The increase of inlet temperature with HPEGR results in a decrease of the RoHR, because of the reduced in-cylinder gas density. Therefore, the introduction of diluents in the combustion chamber results in an increase in the combustion delay as well as a shifting of the combustion process towards the expansion stroke which involves a reduction in the indicated work. This degradation in the indicated work is intensified by the increase in the intake temperature.

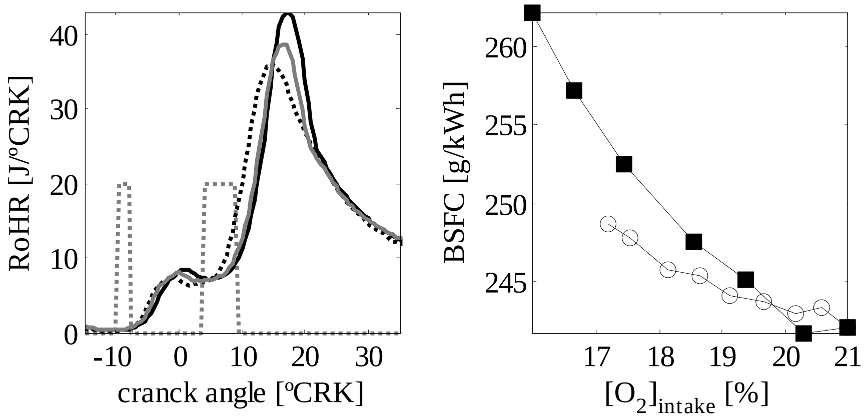

figure 4. Effect of the EGR on the Rate of Heat Release (left) and effect of the intake O2 concentration on the BSFC (right) at A conditions. ···: w/o EGR ([O2]=21%). -: LPEGR ([O2]=17.5%). -: HPEGR ([O2]=17.5%). ···: injection pattern. ○: LPEGR. ■: HPEGR.

To compute the global engine efficiency, the pumping losses, i.e. the work done to evacuate the exhaust gases and introduce the fresh charge into the cylinders, should be taken into account. For a given intake pressure, when a HPEGR system is used, communication between the intake and exhaust manifolds reduces the pressure difference between intake and exhaust manifolds as observed in figure 5 and then pumping losses. The thermal throttling also contributes to decrease the pumping losses since the flow through engine elements and trapped mass are reduced with increasing EGR. When the LPEGR is used, the pumping losses are almost constant due to the fact that the mass flow through engine elements and trapped mass are almost constant. Despite of the higher pumping losses, the higher mass flow through elements with LPEGR presents some advantages, specifically regarding turbocharging, the higher mass flow through the turbine allows the turbocharger to reach a higher speed reducing the turbocharger lag during a sudden increase in load. In addition, the compressor operating points of the engine using the LPEGR system are located close to those of the engine without EGR, while compressor operational points using HPEGR are shifted towards the surge line.

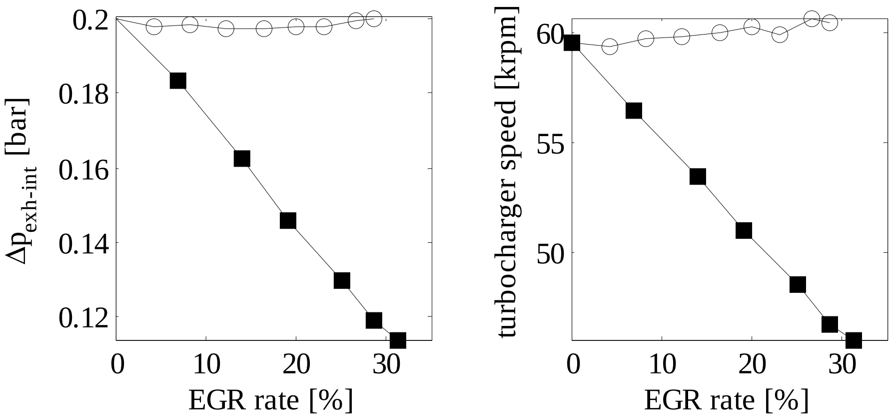

figure 5. Effect of the EGR on the pressure difference between exhaust and intake manifolds (left) and effect on turbocharger speed (right) at A conditions. ○: LPEGR. ■: HPEGR.

Regarding emissions, figure 6 shows the evolution of exhaust gas opacity and NO_x concentration as the intake O_2 concentration is reduced with both LP and HPEGR configurations. As stated before, the main difference between both systems is that while the intake temperature with the LPEGR system has been kept constant (40ºC), it has suffered an important increase (from 40ºC to 115ºC) with the HPEGR loop. When the intake temperature is kept at 40ºC (LPEGR) the reduction in the intake O_2 concentration produces an increase in the exhaust gas opacity up to 14% with a O_2 concentration of 17.5%. From this conditions, an additional decrease in the O_2 concentration leads to a reduction in exhaust gas opacity. According to the literature [13, 14], the lower the intake temperature, the higher the O_2 concentration from which the opacity starts to decrease with EGR. Also, for a given O_2 concentration, the higher the intake temperature, the higher the exhaust gas opacity. This behaviour is reflected in figure 6, where it can be observed how the exhaust gas opacity with the HPEGR system (high temperature) is higher than that obtained with the LPEGR architecture. It can be also noticed that with HPEGR, the reduction in the O_2 concentration involves an opacity increase in the whole tested range.

In the right part of figure 6 it can be seen that for a given O_2 concentration, raising the inlet charge temperature increases the exhaust NO_x concentration. In this way, the lower intake temperature with the LPEGR system allows an additional NO_x reduction to the effect of the O_2 concentration reduction.

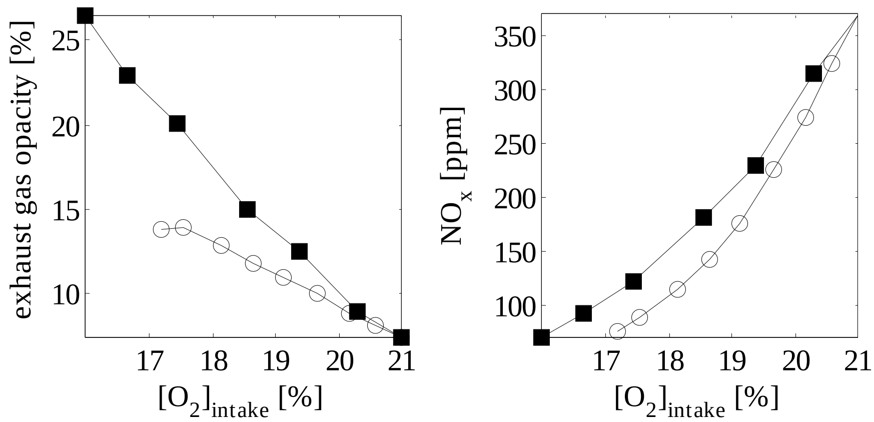

figure 6. Effect of EGR on opacity (left) and NOx concentration (right) of exhaust gases at A conditions. ○: LPEGR. ■: HPEGR.

3.2 HPEGR and LPEGR combination

When the LPEGR and HPEGR systems have been compared, in the present section, the combination of both architectures is analysed. In figure 7, the EGR rate measured at the different tested conditions versus the mass flows through both EGR circuits is represented. For both operating conditions the LPEGR rate is limited by the pressure difference between the DPF outlet and the compressor inlet. The LPEGR rates achieved in this study could be increased by using a backpressure valve in the exhaust line, after the EGR extraction, which increases the pressure difference in the LPEGR line. Nevertheless, this solution will increase the engine pumping losses and will have a negative impact in BSFC. On the contrary, the HPEGR rate is limited by an excessive reduction in the intake O_2 concentration, which involves an important increase in opacity and BSFC.

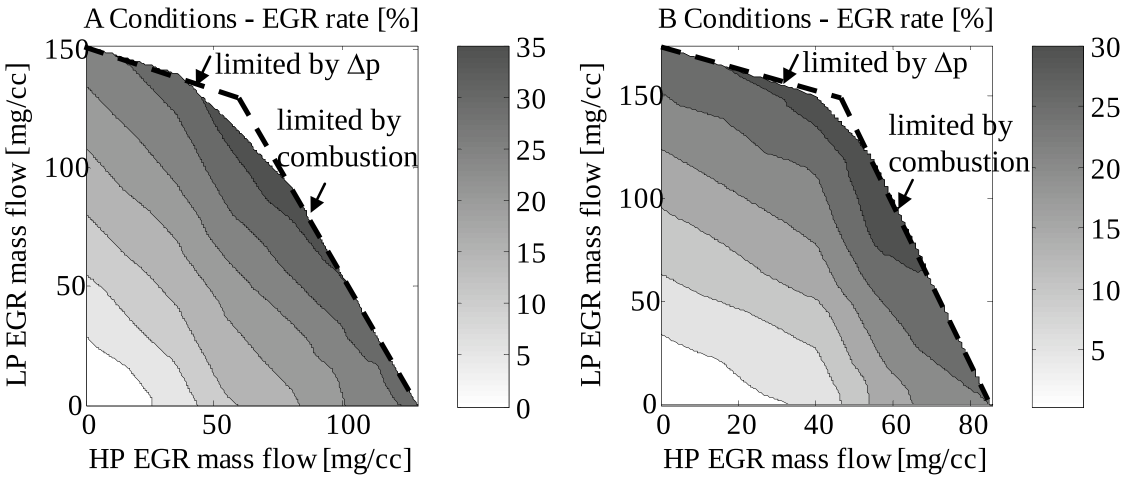

figure 7. EGR rate as a function of the flow through HP and LPEGR Systems at A (left) and B (right) conditions.

The differences in the intake temperature and O_2 concentration lead to differences in the engine behaviour. Results concerning engine performance are shown in figure 8. In this figure the trade off NOx-opacity is represented for both operating points. The color scale represents the BSFC, from dark (high BSFC) to light (low BSFC). The results obtained indicate that for a given EGR rate, the higher the LPEGR contribution, the lower emissions (points are moved towards the origin of coordinates. Also, for both operating conditions, the points with HPEGR show for a given NOx emissions a higher BSFC and opacity.

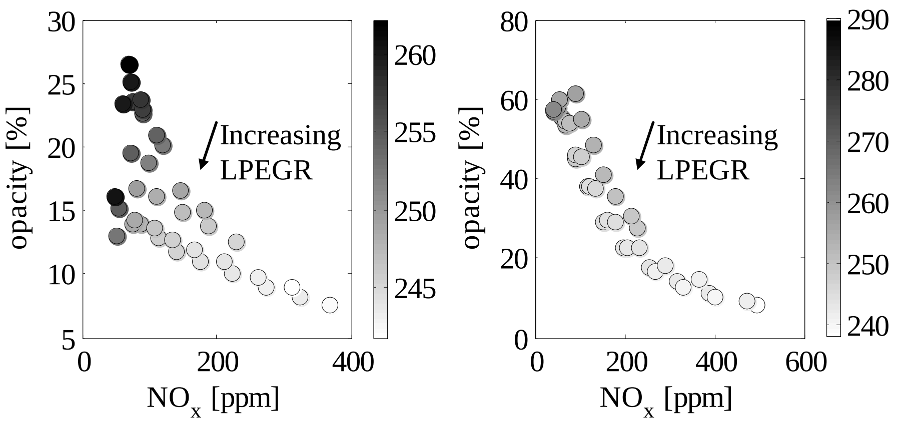

figure 8. Trade off NOx-opacity at A (left) and B (right) conditions. The colorscale represents the BSFC from low (white) to high (black).

In this sense, for the tested conditions the combination of both EGR loops does not improve the performance of the LPEGR system. Nevertheless, other studies are required in order to explore possible benefits of the HP and LPEGR such as:

- The control of both intake charge temperature and composition. In spite of the benefits of the low intake temperature in terms of NO_x and opacity, the low intake temperature also involves higher HC emissions, especially during the warm up. In this sense, figure 9 shows the effect of the EGR on HC emissions during the NEDC, it can be observed that especially during the first phases of the cycle, where the engine is still cold, the HC emissions with the LPEGR system are considerably higher due to the lower intake temperature. As the cycle evolves and the engine warms up, these differences are progressively reduced. Since the benefits of the LPEGR configuration on NO_x mainly appear at the last phase of the NEDC, the HP-LPEGR combination allows taking profit of the lower HC emissions of the HPEGR configuration during the ECE, while reducing the NO_x emissions during the EUDC due to the use of the LPEGR architecture.
- The increase in the EGR rate without increasing pumping losses avoiding the use of a backpressure valve. In fact, the HPEGR can be used to increase the LPEGR rate up to the required levels.
- Avoiding high exhaust gas concentrations through the intake line that can produce condensation.
- At high speed and loads, where the mass flow is high and therefore pumping losses become important, the substitution of the LPEGR rate by HPEGR can improve BSFC.

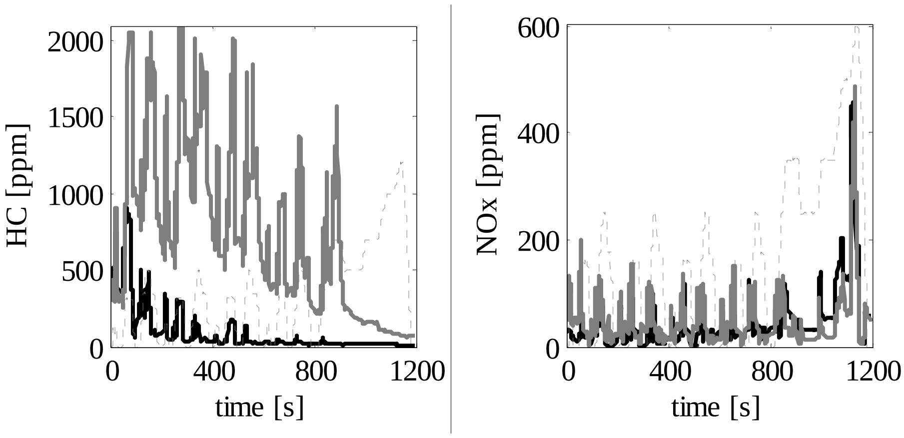

figure 9. Effect of EGR on HC (left) and NOx (right) emissions during the NEDC. -: LPEGR. -: HPEGR.

4. Conclusions

In the present paper the effect of both LP and HPEGR architectures on engine performance and emissions has been analysed. The following conclusions have been obtained:

- The LPEGR loop allows increasing the EGR rate without increasing the intake temperature. The increase in intake temperature promoted by the HPEGR leads to a reduction of intake gas density and therefore on trapped mass. In this sense, the HPEGR reduces the intake O_2 concentration by substituting the fresh air (dilution effect) and also by reducing the trapped mass (thermal throttling).
- Due to the thermal throttling effect of HPEGR, for a given EGR rate, the intake O_2 concentration obtained with the HPEGR system is lower than that obtained with the LPEGR configuration.
- The reduction in the mass flow through intake and exhaust lines with HPEGR gives rise to a reduction in the pumping losses as the EGR rate increases (keeping constant intake pressure). Nevertheless, this reduction also involves a lower mass flow through the turbine and then less turbocharger speed. The lower turbocharger speed damages the engine transient performance during a tip in.

- The increase in intake temperature with higher HPEGR rates has a negative effect on BSFC, NOx and opacity.
- For the conditions evaluated in this study, the combination of LP and HPEGR systems does not involve any benefit comparing with the BSFC, NO_x and opacity obtained with the LPEGR system.

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POTENTIAL OF USING A NOZZLE AT THE COMPRESSOR INLET OF A HIGH SPEED DIRECT INJECTION DIESEL ENGINE

Abstract

The effect of a nozzle placed upstream the compressor of a High Speed Direct Injection (HSDI) Diesel engine is discussed in this paper. It is observed that some engine operation points, which are placed into the surge area of the compressor, become reachable due to the effect of the nozzle in the compressor behaviour. The nozzle has been characterized in a flow test rig in order to determinate its behaviour up to sonic conditions. The operation map of the set nozzle-compressor, working as a unit, has been obtained in a turbocharger test bench. The distortions in the compressor inlet are usually considered as a disadvantage, because they affect the efficiency and the behaviour of the turbocharger. However, in this paper the comparison between the original compressor map (provided by the costumer) and the nozzle-compressor map, shows that the distortion caused by the nozzle improves the engine-turbocharger coupling, then reducing, for a given pressure ratio, the air mass flow which leads to surge in a 30%.

The improvement in turbocharging may not increase the performance of the engine. Nevertheless, the possibility of increasing the boost pressure allows increasing the fuel supply, having properly combustion process and therefore maintaining efficiency, then increasing the power of the engine.

Key words: Turbocharging, surge, diesel engine, downsizing

1. INTRODUCCION.

1.1 Objective and Methodology.

The aim of this study is to analyse how the use of a nozzle placed at the compressor inlet affects the compressor behaviour. Particularly, it is studied how the nozzle affects the surge line and the compressor efficiency. The reduction of the surge area produces an increase of the compressor operation range without diminishing its efficiency, discounting the nozzle losses.

In order to study the specific behaviour of the nozzle, it was tested and characterized in the test rig. Then, the compressor with the nozzle was tested in a turbocharger test bench in order to obtain its operation map. Once the new map was obtained, it was compared with the compressor map supplied by the manufacturer to show the differences and to evaluate the benefits of using an inlet nozzle.

1.2 Turbocharger range operation.

The use of turbocharging in engines is a widely used technique to obtain higher power rates. Besides, turbocharging allows increasing the engine efficiency. Despite being highly developed, the research in turbocharging is still relevant today. Downsizing is a current trend in engine development, which objective is to obtain higher power rates and better engine efficiencies. In this sense turbocharging is suitable technique to allow downsizing. In addition to the economical effects of the reduction in the fuel consumption, this drop entails a decrease in the CO2 emissions. The political measures and the social demands regarding CO2 emission make the downsizing an interesting research line for both Gasoline and Diesel engines [1]. One of the most effective ways to allow the engine's downsizing is to employ high boosting in the whole range of the engine operation.

The compressor surge and choke limits are one of the main problems in downsizing, because both reduce the engine operation range. For a given engine, turbochargers, which guarantee high pressure at low mass flows, do not use to be able to supply the required air mass flow at high load-speed engine conditions due to compressor choke. On the contrary, when a turbocharger able to feed the engine with the required air mass flow at high load-speed conditions is employed, there is a risk of reaching surge when the engine operates at low load and speed conditions. Then, the intake pressure at these conditions should be reduced to avoid surge, which involves a decrease in engine performance. In summary, the turbocharger-engine matching is an important problem to be faced by current engine designers.

Several technical solutions have been developed to increase the operation range of the turbocharger. Improvements in the impeller design, the use of diffusers in the compressor outlet or twin turbocharger systems are some of the approaches to increase the compressor range. In [2] a further revision of these systems is presented.

1.3 Effect of the compressor inlet conditions in the surge line. State of the art.

A revision of the state of the art concerning the effect of the compressor inlet conditions on surge has been carried out. The studies found, are aimed to determine the losses and the problems of the different shapes and geometries of the compressor inlet pipes. Due to the engine packaging requirements, the compressor inlet pipes usually have a strong bend (90º) just before entering the compressor. The pipe bending not only increases the friction losses but also produces an asymmetry in the flow conditions. As the flow cannot be stabilized before the impeller, the asymmetric flow affects the compressor behaviour, as can be seen in [2,3,4,5,6]. Many of these researches expect to develop models which will be able to simulate this behaviour. In [5] a swirl system is proposed to reduce this asymmetry conditions.

An important conclusion of these researches is that a restriction in the compressor inlet causes a displacement of the surge line to the left of the compressor map, increasing the achievable pressure ratio at low air mass flows.

In [3,7] flow distortions with radial symmetry, such as the nozzle presented in this paper, are studied. In [7] the surge phenomenon in a compressor for aeronautic applications is carefully analyzed. Also, several configurations are described to increase the achievable pressure ratio for low air flow demands avoiding surge. One of those configurations consists of a compressor with variable inlet vanes to improve the starting process of an aeroengine. During the starting process of an aeroengine, the airflow remains low (the aeroengine is stopped) and high pressure is needed to achieve the starting torque. Nevertheless, this system is not used in automotive as a technique to avoid surge.

In this line, at [8] a variable inlet guide vanes with a pre-swirl generator is tested to improve the operating range of the compressor. In this study it is expected to increase the stable air flow range at high pressures ratios, displacing the surge line as left as possible in the compressor map.

2. ENGINE-TURBOCHARGER MATCHING

The maximum power which can be obtained from an engine is limited by the fuel amount can be burned into the cylinders. The combustion will be suitable if there is enough air in the combustion chamber. The air amount needed to have an efficient combustion and the engine properties (volumetric efficiency, ηv) define the requirements of both engine mass flow and intake pressure [9]. When the engine conditions are defined, a selection of the turbocharger must be done among all the turbochargers able to achieve this operating point.

Although the objective of this paper is not the powering of a particular engine, the improving of the compressor behaviour allows increasing the power of turbocharged engines working at some operating conditions.

For a given engine operation conditions, the intake pressure and the mass flow rate can be calculated by using the power and volumetric efficiency equations.

$$N_e = \eta_e \cdot m_a \cdot F \cdot H_c \quad (1)$$

$$\eta_v = \frac{\dot{m}_a}{\frac{1}{2} \cdot n \cdot V_t \cdot \frac{P}{R \cdot T}} \quad (2)$$

Where the efficiency (η_e), volumetric efficiency (η_v) and the displacement (V_t) are inherent characteristics of engine, the fuel-to-air ratio and the engine speed are imposed and, finally, the intake temperature depends on the characteristics of the intercooler and intake pipes.

In addition to the previous parameters that define the turbocharger behaviour at steady conditions, the turbocharger inertia should also be taken into account to ensure a suitable transient performance.

As an example, in fig. 1 shows the compressor maps for two different turbochargers, A and B. Both of them are able to provide the required mass flow and pressure ratio to reach the target operating point. On the one hand, the operating point is placed at the maximum efficiency area of compressor A. On the other hand, the operating point is located at the boundaries of compressor B. According to the maps, geometrical parameters of compressor B should be smaller than those of compressor A. For this reason, the inertia of compressor B will be lower and it will have better transient response than compressor A, which involves advantages for some applications.

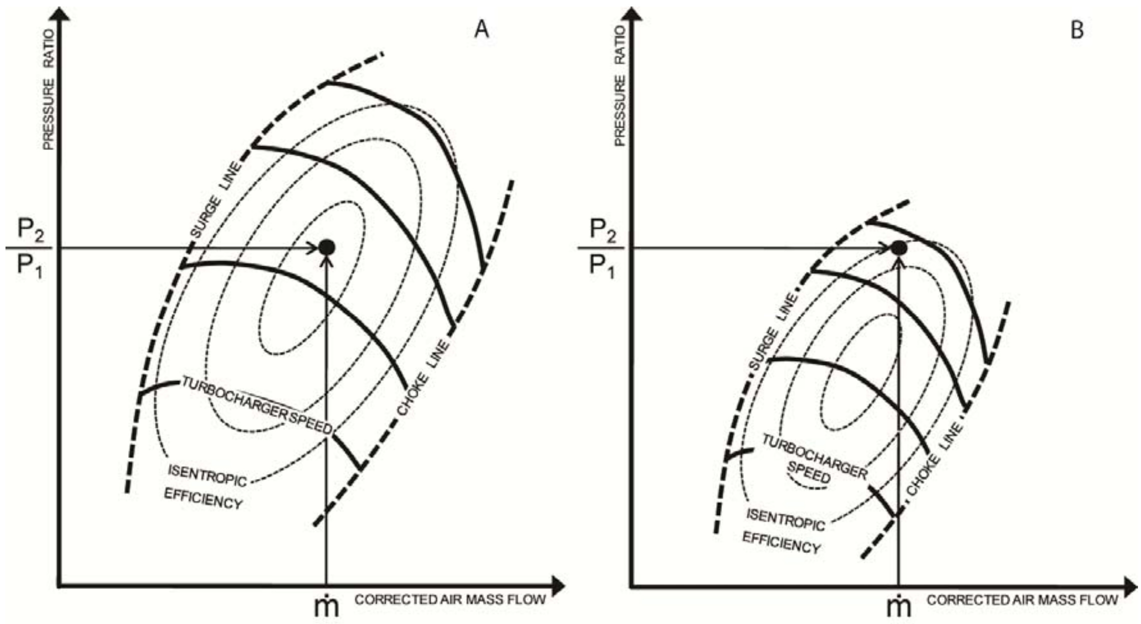

Fig. 1 Compressor maps, which both of them achieve the operation point with different inertia.

The map of the compressor used in this work, was provided by the manufacturer, as fig. 2a presents. The normal compressor operating range can be extended to the surge zone by installing a nozzle at the inlet of the compressor, as fig. 2b shows. It can be observed that the working conditions, represented by the black star (), are placed in the surge area of compressor 2a, whereas the same conditions are achievable with compressor 2b.

1- In order to avoid surge, three different approaches can be followed:
2- Decrease the pressure ratio, which entails a decrease in engine performance.
3- Change the current turbocharger for a larger one, giving rise to a lower transient performance due to the higher inertia.

Modify the inlet geometry to reach the required operating point, which can comprise the compressor efficiency as the solution adopted in this work.

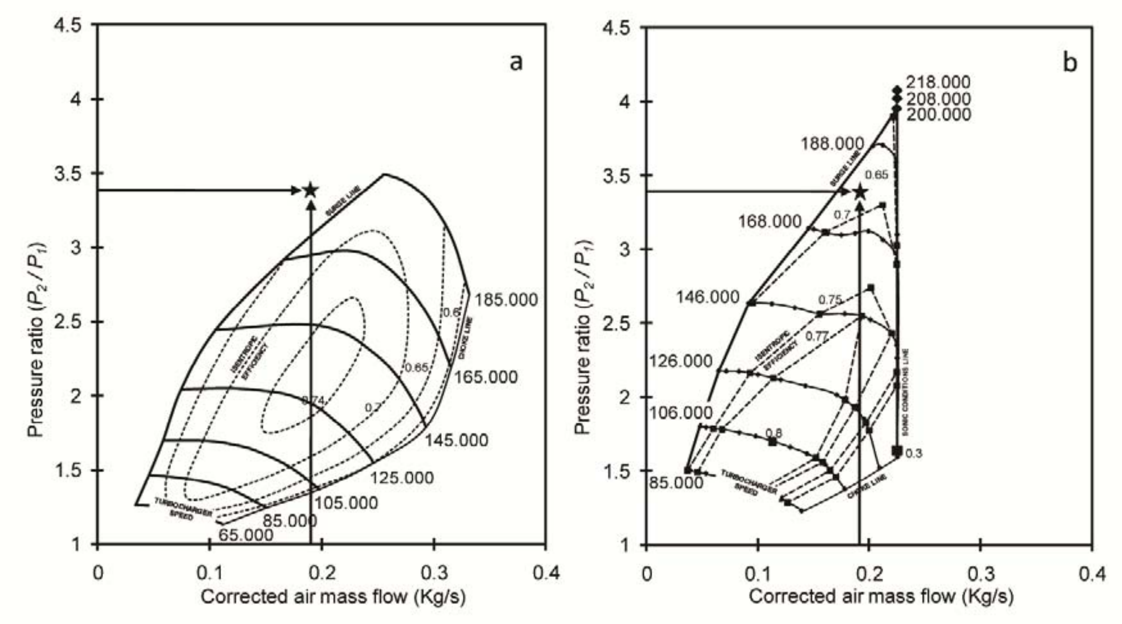

Fig. 2 Original compressor map with the desired operation point (a). Measured Map of the Compressor working with the nozzle (b).

3. NOZZLE STUDY

Before the turbocharger test, the nozzle was analyzed individually. The main objective of this analysis was to characterize the nozzle in a test rig and to check its real behaviour. Then, this characterization is compared with the isentropic behaviour to evaluate the nozzle design.

The nozzle throat is 35 mm in diameter and 3 mm in length and it is located in the compressor inlet 50 mm before the impeller. The nozzle profile has been designed to minimize losses with a convergent-divergent shape. The convergent duct guides the air flow to the nozzle throat, maximizing the nozzle effective section. However, this guide has to minimize the friction losses reducing the length of this part. The divergent duct should transform the flow kinetic energy in pressure energy as isentropically as it can.

In this sense, the nozzle profile has been designed by means of Hermite polynomials, which describe the trajectory between two points with the lower curvature for a given maximum derivative. The use of this polynomial guarantees the minimum surface in contact with the air [10].

The sketch nozzle is represented in fig. 3.

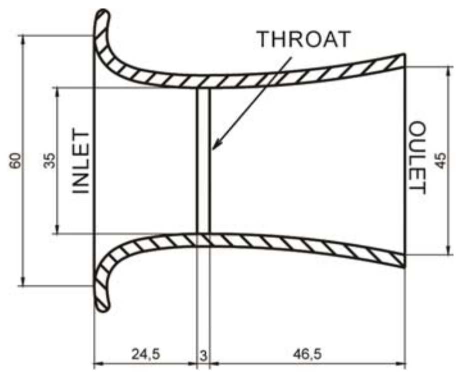

Fig. 3 Nozzle sketch and picture of the used nozzle.

The flow through the nozzle is limited by the sonic conditions at the throat (Mach=1). These conditions can be calculated by considering steady compressible flow and supposing an isentropic evolution in the nozzle [11], according to the following equations:

$$\dot{m} = A_1 \cdot c_1 \cdot \rho_1 = A_2 \cdot c_2 \cdot \rho_2$$ Continuity equation (3)

$$h_1 + \frac{c_1^2}{2} = h_2 + \frac{c_2^2}{2}$$ Energy equation (4)

$$\frac{P_1}{P_2} = \left(\frac{T_1}{T_2}\right)^{\frac{\gamma}{\gamma-1}}$$ Isentropic evolution equation (5)

Therefore, for the initial conditions given in the nozzle test:

P_{intake}=1022 millibar

T_{intake}=22ºC

By using previous equations, the maximum mass flow that can go through the nozzle in an isentropic evolution can be calculated, obtaining 0.234 Kg/s.

3.1 Nozzle test rig

The flow test rig, which is represented in fig. 4, consists of a Roots compressor which gives continuous air flow, a hot film anemometer sensyflow to measure the mass flow, a settling tank of 200 litre volume in order to ensure stagnation conditions downstream of the tested devices and pressure sensors. The pressure in the settling tank (P0) and in the nozzle throat (PThroat) has been measured by means water level sensor. A deeper description of the test rig can be found in [12].

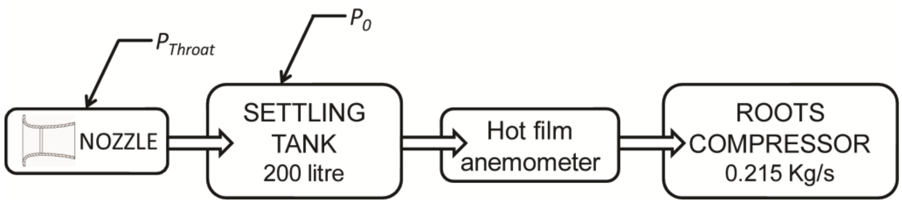

Fig. 4 Test rig layout.

The Roots Compressor produces a suction which induces the air flow from the atmosphere to the settling tank where stagnation conditions are reached. A 200 litre tank guarantees steady conditions after the tested element and also before the hot film anemometer to allow a suitable air mass flow measurement. The nozzle is connected to the settling tank by a pipe of 10 diameter length, where the reference is the diameter at the end of the divergent pipe. This assembly prevents aerodynamic phenomena inside the nozzle (turbulence, choke,…) from affecting the stagnation conditions in the deposit.

In the test, for a given air mass flow, the pressure differences between the ambient and the deposit and between the ambient and the nozzle, are measured by means of a liquid column manometer. The test is started with 0.085 Kg/s and it is increased until the maximum mass flow that can be moved by the roots is reached (0.215 Kg/s) by means of 0.015 Kg/s steps.

3.2 Test results.

The nozzle curves, obtained in the test, are represented in fig. 5. The black line shows the mass flow versus the expansion pressure ratio measured in the nozzle throat. The grey line represents the calculated isentropic evolution. The flow equation (6) has been used to estimate the non-isentropic evolution.

$$\dot{m} = K\sqrt{2 \cdot \Delta P \cdot \rho} \quad (6)$$

Where $\dot{m}$ is the air mass flow, ΔP is the pressure between the atmospheric and nozzle throat pressure, ρ is the air density and K is a correction.

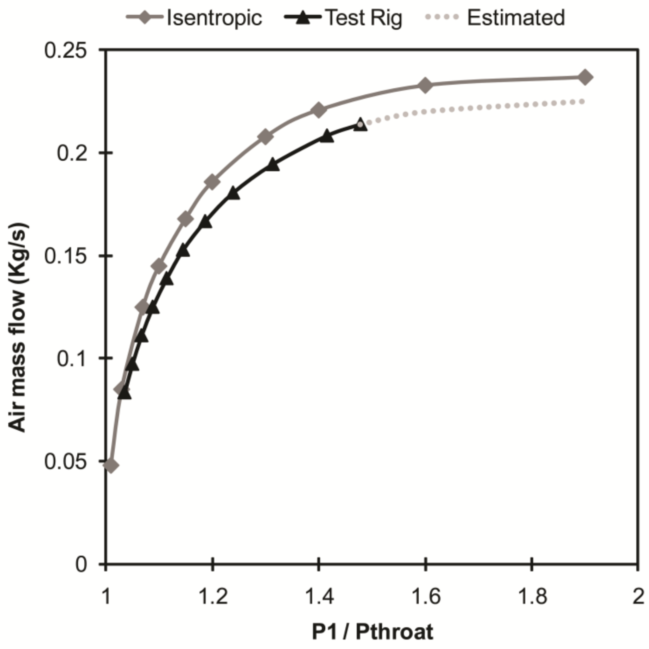

Fig. 5 Nozzle characteristic curve

The difference between the real mass flow and the one calculated by the isentropic equations can be clearly observed. This difference is small for low mass flow, but it increases as the mass flow rises. This fact is due to the air friction and the heat transfer in the nozzle wall. When the air velocity increases near to sonic conditions, the air pressure falls and its temperature decreases. In this case -20ºC can be reached. The heat transfer between the air and the nozzle wall, which is near at atmospheric temperature, reduces the isentropic behaviour of the process. In any case, this effect was taken into account when compression efficiency was calculated.

4. TURBOCHARGER TEST

4.1 Turbocharger Test Bench.

The characterization of the compressor working with the nozzle has been carried out in the CMT turbocharger test bench [13].

The experimental facility, which is represented in fig. 6, is based on a heavy-duty diesel engine, used as a hot gas flow rate generator to feed the turbine/turbocharger is going to be tested. This engine has 6 cylinders with 11 litre of displacement and it is connected to an electromagnetic brake. A screw compressor is used to feed the engine with the required air mass flow, while the engine exhaust gases are used to drive the turbine (a Variable Geometry Turbine which was set in a constant position). The limits for the test bench are given by the maximum air flow that can be displaced by the screw compressor (0.5 kg/s) and the maximum temperature can be reached safety in the exhaust line (750 ºC).

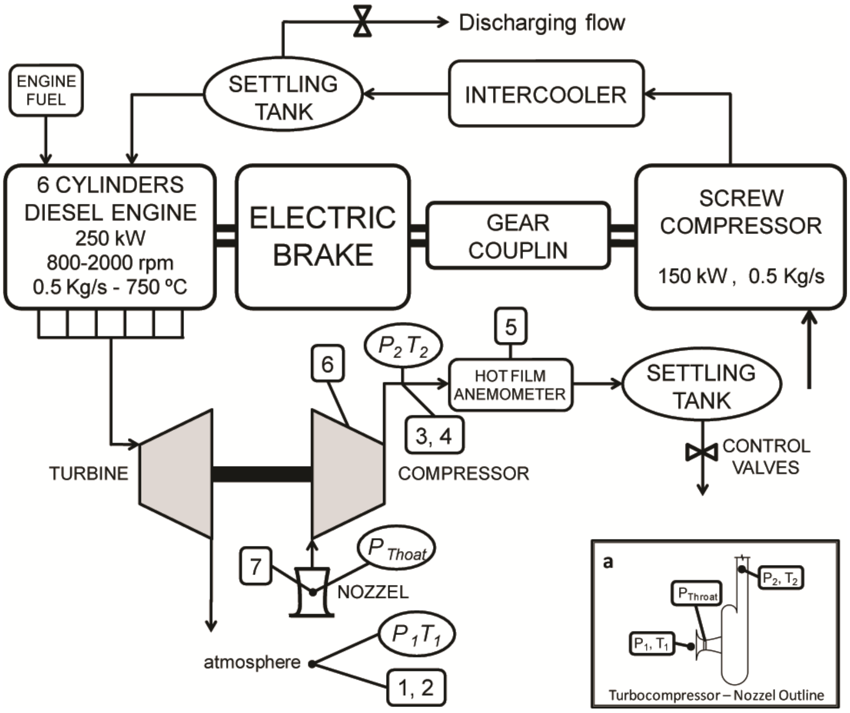

Fig. 6 CMT Turbocharger test bench layout.

The tested compressor takes the air from the test room directly. The mass flow that passes through the compressor is controlled using a control valve placed at the compressor outlet.

The compressor map can be obtained by means of the system instrumentation. The used sensors, which are also numbered in fig 6, are:

1- Compressor inlet pressure in stagnation conditions measured by a piezoresistive pressure sensor.
2- Compressor inlet temperature measured by a K type thermocouple.
3- Compressor outlet pressure. It is measured by a piezoresistive pressure sensor.
4- Compressor outlet temperature. It is measured by a thermocouple K.
5- Air mass flow. It is measured downstream the compressor with a Sensyflow hot film anemometer supply by kistler.
6- Turbo group rotating speed measured by an eddy current sensor.
7- Nozzle throat pressure.

The sensor characteristics are detailed in table 1.

Table 1. Turbocharger test bench sensors

	Pressure	Temperature	Air Mass Flow
Model	Kistler 4045 A5	K	Sensyflow P DN 150
Range	0 - 5 bar	0 - 1260 ºC	80 - 2400 kg/h
Precision	0.1	0.3	1.5

4.2 Turbocharger test procedure.

The test conditions and the method used are described in the following points:

1. The tested turbocharger rotating speeds range from 85000 rpm to 200000 rpm. Measurements were done at 20000 rpm intervals.
2. For each rotating speed, the air mass flow ranges from the flow that produces surge to that obtained in choke conditions. For a given turbocharger speed, the air mass flow is reduced by closing the compressor outlet valves. Choke conditions are produced when the valves placed at the compressor outlet are fully open and the flow through the compressor becomes maximum. The surge conditions are detected by the flow fluctuation, created by the instable behaviour of the compressor according to the procedure presented in [14]. Reference [14] presents an algorithm based on the frequency analysis of the instantaneous pressure in the compressor outlet that allows defining when unsteady behaviour is achieved. The pressure fluctuation measured during surge phenomenon is associated to air flow fluctuations.
3. The measurement obtained in the test has been corrected by the pressure and the temperature from the compressor inlet. In order to take into account the influence of ambient conditions, the same methodology used by the turbocharger manufacturers has been used. The correction equations, in IS units, are the next:

$$N_c = \frac{N_{phy}}{\sqrt{T_1/302.78}} \ (7); \quad W^* = \frac{W\sqrt{T_1/302.78}}{P_1/96173} \ (8)$$

- N_c is the corrected turbocharger speed and N_{phy} is the measured.
- W^* is the corrected air mass flow and the W is the measured air mass flow.

If the test conditions are used, magnitudes of the corrections are:

$$N_c = 1.005 \cdot N_{phy} \ (9); \quad W^* = 0.94 \cdot W \ (10)$$

4. The isentropic efficiency represents the comparison between the work of the actual compression and the work in an isentropic compression. This parameter can be calculated as [13] :

$$\eta_{isentropic} = \frac{h_2 - h_1}{h_{2s} - h_1} \ (10)$$

$$h = C_p \cdot T \ (11)$$

$$\eta_{isentropic} = \frac{T_2 - T_1}{T_{2s} - T_1} \ (12)$$

Where T_{2S} is calculated as isentropic compression:

$$\frac{T_1}{T_{2s}} = \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} \ (13)$$

5. The nozzle has been maintained during the whole test.

4.3 Results from the turbocharger test with a restriction nozzle.

Both the new compressor map obtained from the test and the original compressor map, which has been provided by the manufacturer, have been compared. Fig. 7 presents this comparison.

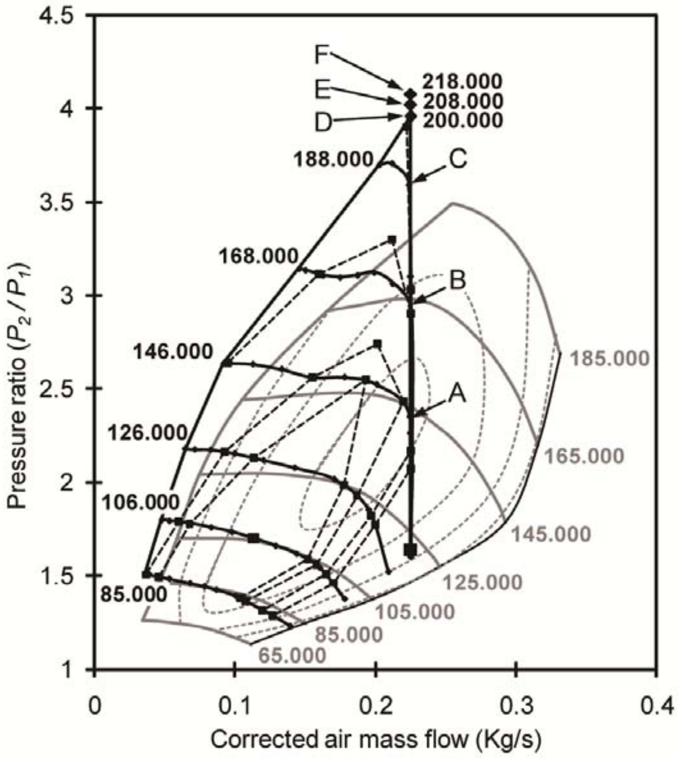

Fig. 7 Comparative of the both compressor maps

The first important difference between both maps is that the nozzle moves the surge line to lower mass flows. It means that for the same air mass flow near the surge line, higher pressure ratios can be reached.

Another fact is that, for low mass flows, the use of the nozzle moves the iso-speed lines to higher pressure ratios. Whereas, for low rotating speed and high mass flow region, this tendency is reversed.

On the other hand, despite the fact that the compressor is working with a restriction at its inlet, the compressor efficiency is kept close to its original values for a wide zone of compressor operating conditions. Fig. 8 shows the isentropic efficiency of the compressor working with the restriction nozzle.

It is observed that, for all compressors rotating speed the maximum compressor efficiency remains almost constant, with values near to 0.8. However, the compressor efficiency falls drastically when sonic conditions at the nozzle are reached. The sonic conditions at the nozzle involve the air mass flow remains constant, independently of the compression ratio. In these specific conditions, the streamlines do not follow the compressor wheel geometry and the compressor efficiency falls drastically.

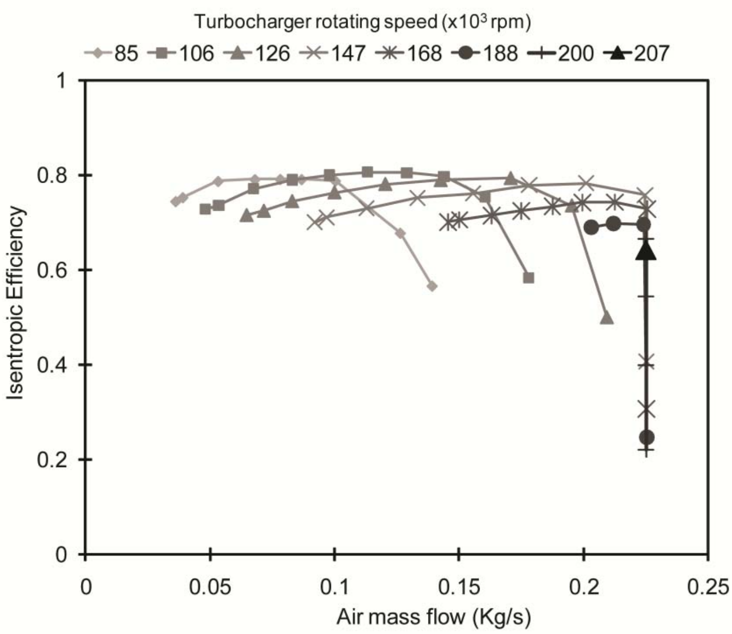

Fig. 8 Efficiency of the set nozzle-compressor

Again in fig 7, the effect of the sonic conditions in the nozzle throat on the compressor behaviour can be noticed. A vertical line marks the mass flow limit of the compressor. Despite the increase in the turbocharger speed, the air mass flow through the compressor is kept constant due to the nozzle choke. Despite this limit, for a constant rotating speed, the compressor can give different pressure ratios depending on the compressor outlet valves position. This effect can be seen in fig. 9. Fig. 9 shows the maximum and minimum pressure ratio that can be achieved for a constant turbocharger speed when sonic conditions are reached in the nozzle. It can be seen how minimum pressure ratio (1.6) remains constant at any turbocharger speed.

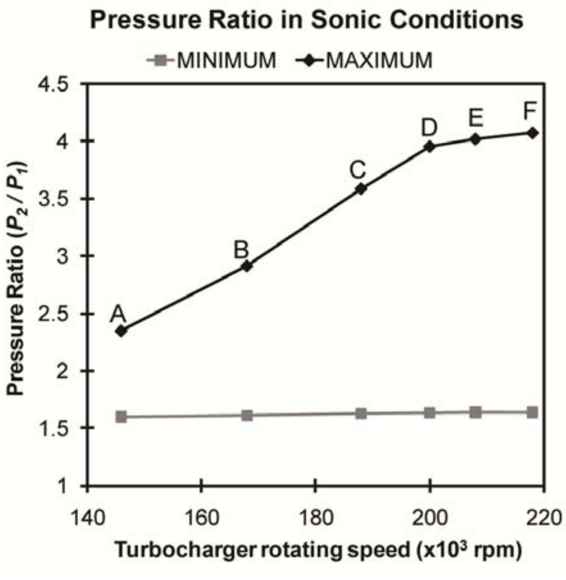

Fig. 9 Pressure ratio when the sonic conditions are in the nozzle.

Another aspect that can be observed at the top of fig 7 is that the line of maximum air flow crosses the surge line. In order to extend the field of the compressor map, points E and F was obtained by increasing the rotating compressor speed, maintaining maximal mass flow rate. A significant rotating speed increase, above 200000 rpm, increases neither the mass flow nor increases the pressure ratio. It is observed that the rotating speed does not increase in a proportional way to the pressure ratio, as can be seen in fig. 9.

5. ENGINE APLICATION

The study made in this paper can be used for improving a high performance HSDI Diesel engine for racing applications. Traditionally, Diesel engines were not used in competition, except for some categories (trucks), due to the fact that, up to now, for a given engine displacement, Diesel engines have a lower effective power and show a slower transient response than spark ignition engines.

The effective power can be obtained by increasing the engine speed or the BMEP. In the SI engine high effective power is obtained raising the engine speed, in this way, the engine is optimized to work at a high rotating speed. However Diesel engines have a very limited rotating speed due to the relative slowness of the mixture and combustion process.

Nowadays, the cutting-edge Diesel technology has allowed this kind of engines to be competitive in a field traditionally dominated by spark ignition engines such as racing cars. This is possible thanks to the high boost levels applied and the new systems of high pressure fuel injection. They improve the mixture and combustion process and allow obtaining a high BMEP from the engine. Therefore, a higher effective power is obtained in Diesel engines with lower engine speeds than those reached in spark ignition engines.

The tested nozzle and the turbocharger were installed in a 2.0 litre HSDI engine with common rail injection.

The final objective of this application was to increase the engine power in the whole operation range by using a larger turbocharger. In order to meet the target power of engine, high boosting is required, also maintaining the compressor stability.

The actual engine intake conditions, pressure ratios and corrected air mass flow, measured in a test bench for different engine speeds are shown in fig 10 (a and b). Thus, the thick line in both graphics in fig 10 represents the engine operation points at full load. In fig 10a the engine operations points are drawn together with the original compressor map. It is possible to observe how the engine operation points would not have been achieved, because they would have been in the surge area.

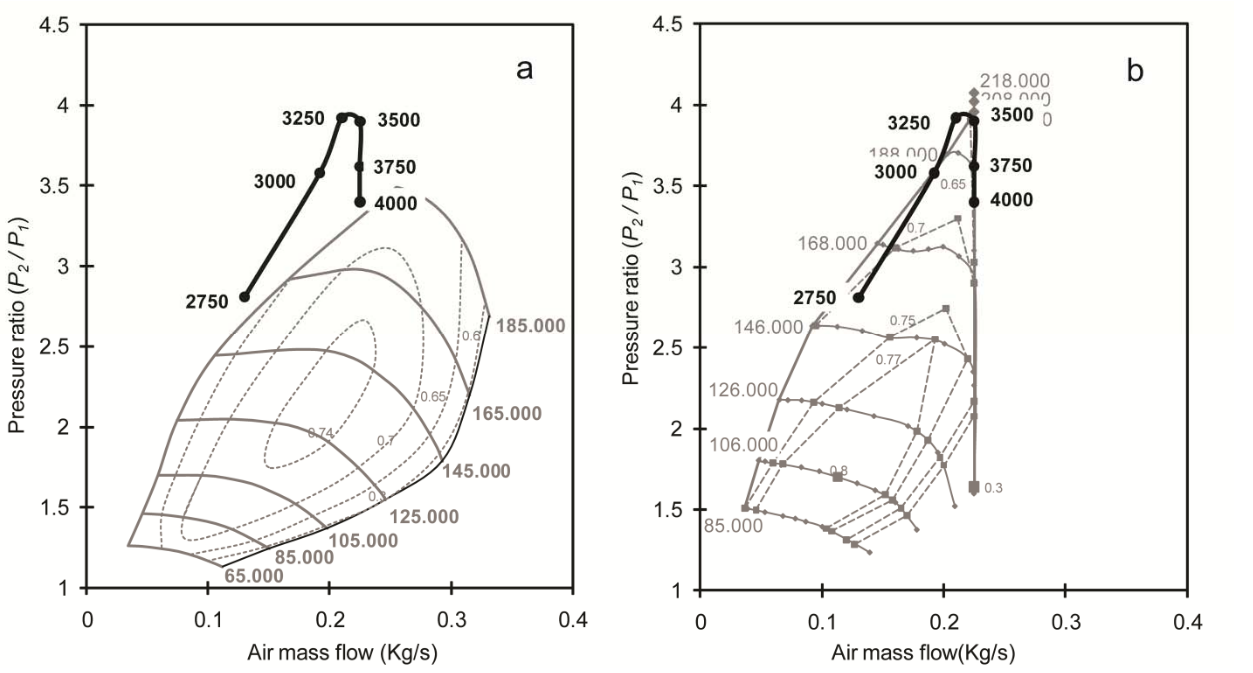

Fig. 10 Engine operation points on the Manufacturer compressor map (a). Engine operation points on the measured compressor map (b).

On the other hand, in fig 10b, where the operation points are represented in their actual compressor map (with the inlet nozzle installed), it is possible to observe how the compressor surge area moves on the left side allowing the required boosting conditions are placed out of the compressor surge zone.

The improvement of this application deals with how a given compressor can be used in order to achieve high pressure ratio operation points, which are out of its original compressor working area. The increase of the compressor pressure ratio in low flow range, which means higher air density, allows increasing the engine fuel supply with convenient combustion efficiencies, which involves directly increasing the engine power expanding the operation range for low engine speeds.

6. CONCLUSION

In the present paper, the effect of a nozzle placed upstream the compressor of a HSDI Diesel engine on the turbocharger has been analysed. A nozzle has been designed by means of a CFD code to reduce its losses as much as possible. The nozzle has been build and characterized in a test rig. The results obtained in the test rig show that the behaviour of the designed nozzle is almost isentropic. The nozzle has been installed at the compressor inlet and the map of the set nozzle-compressor has been measured in a turbocharger test bench. Finally, by means of the comparison between the original compressor map and the map obtained when the nozzle is employed the following conclusions are obtained:

1- The restriction in the compressor inlet produced by the nozzle entails a reduction of the surge area. This effect involves the possibility of obtaining higher pressures ratios for lower mass flow.

2- The nozzle-compressor set has almost the same isentropic efficiency than the compressor. It means that the isentropic nozzle loss is not important, due to the good nozzle design.

3- There is a limit in the compressor mass flow that is reached when sonic conditions appear in the nozzle throat.

4- The most important contribution of this study is the application of the nozzle effect on the compressor to a real engine. The stability of the compressor behaviour, maintaining high pressures with low flow rates, enables the engine to be highly boosted at low engine speeds. In this conditions, it is possible to increase the power of the engine only by increasing the fuel supply

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