Lecture 10

3rd Semester M Tech. Mechanical Systems Design

Mechanical Engineering Department

Subject: Advanced Engine Design

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Topic: Design Of A Heavy Duty Truck Engine

A Turbocharged Compression Ignition Engine Design

Objective: Estimate Engine Displacement Volume Required – 05-10-2020

Numerical Example:

Q1 Design a Heavy Duty Truck Engine
The engine is to have a rated power of 300 KW at 1800 rpm.

Solution:

The engine has to be designed for a passenger car run by diesel fuel
This is a C.I engine based Passenger Car

Given Data:

Rated Power = 300 KW

Rated Speed = 1800 rpm

The above Rated Speed suits a heavy duty truck engine for the following reasons:

Consider the equation for **Newton's Second Law of Motion** for a reciprocating **piston**.

F = m*a

Reasons for low rated speed of 1800 rpm

Reason 1

Lean Operation of CI engines.

A/F = 18 - 70

1. For pollution control the CI engines are operated **lean.**
2. So CI engines have larger displacement volume as compared to SI engines even for same power.
3. Larger displacement volume will make the **geometrical size** and therefore the **mass of the piston** for CI engine larger.
4. This dictates a need for decreasing the crankshaft rotational speed.

Note:
Inertia is associated with mass only. It means resistance to motion offered by the mass m of the piston against the force F acting on the piston.
At this point we are not talking of the combustion based piston motion.
Rather we are talking about the force F and the corresponding resistance offered by a mass m of the piston to this force F.
Such analysis can be done by using external motors driving the crankshaft and therefore the piston moves up and down without combustion In the engine cylinder.

Reason 2

Pressure ratio ($\frac{P2}{P1}$) across the compressor of the turbocharger

1. **A heavy duty compressor** is incorporated in the turbocharger of **a heavy duty truck engine**. The much **higher pressure ratio** of the compressor compresses the **air** to a much **higher density** before sending it to the engine cylinders. This in turn results in a **large force F** to act on the piston of a heavy duty truck engine during **combustion** process.
2. Since the size and therefore the mass m of the piston on the right hand side of the Newton's second law of motion is also on the higher side for such a heavy duty CI engines, so the second term acceleration a of the piston has to be brought down accordingly.
3. This finally dictates for the design consideration of a lower crankshaft rotational speed N.

Reason 3

Lower B/L ratio for CI engines

4. The B/L ratio for CI engines is lower than that for SI engines.
5. Therefore the CI engines have longer stroke than SI engines in order to maintain a high combustion efficiency with heterogeneous combustion.
6. Longer stroke L of the cylinder will increase the value of acceleration or deceleration a of the piston between TC and BC for the same crankshaft rotational speed N.
7. **In order to keep the inertia force F same, the crankshaft rotational speed N needs to be brought down.**

Therefore crankshaft rotational speed was chosen as:

N = 1800 rpm

We Know - The best possible Brake Specific Fuel Consumption for C.I engine = 200 g/KWh

Let

BSFC = 210 g/KWh

Reasons –

Refer the chapter: characteristics Performance Curves of C I engines under variable speed operation

1. From the curves we see that the best possible BSFC or minimum BSFC is towards the speed for maximum torque
2. We get best possible combustion efficiency at this engine speed which minimizes the fuel consumption.
3. On the same curve for BSFC when we go towards the idle speed or towards the rated speed for maximum power – the BSFC increases in both directions with a corresponding drop in combustion efficiency
4. The BSFC for the turbocharged heavy diesel engine is lower than that of a naturally aspirated diesel engine since approximately 9.6% energy is recovered from the exhaust gas by incorporating the turbine of a turbocharger. This decreases the fuel consumption per unit energy output.
5. The lower crankshaft rotational speed of the engine as dictates by the preceding discussion, will further reduce the number of power cycles per unit time. This is turn reduces the mass flow rate of the fuel to the engine cylinders. This lower mass flow rate of the fuel also results in low numerical value of BSFC of the engine.

Let

Volumetric Efficiency = 91 percent

(when intake port is a reference)

Volumetric efficiency = 279 %

(when ambient air is a reference)

Reasons:

1. When intake port of the engine cylinders is taken as a reference the volumetric efficiency of the cylinders of a heavy duty truck engine is 91 percent.
2. When ambient air is taken as a reference for the cylinders of a heavy duty truck engine the volumetric efficiency of the engine increases beyond 100 percent.
3. A heavy duty compressor in a heavy duty turbocharger for a heavy duty truck engine increases the volumetric efficiency of the engine to a much higher value.
4. A heavy duty compressor is used to increase the density of the ambient air to as much as 3 times before its entry to engine cylinders.

By using the equation for the definition of BSFC

BSFC = $\frac{\dot{m}_f}{P}$

Where

$\dot{m}_f$ = mass flow rate of fuel

P = Power developed by the engine

Or

$\frac{\dot{m}_f}{P}$ = 210 g/KWh

Power = 300 KW

(The design of a heavy duty truck engine)

Therefore substituting above:

$\dot{m}_f$ = 210 g/KWh * 300 KW

$\dot{m}_f$ = 63,000 g/h

$\dot{m}_f$ = 1.05 Kg/min

Mass flow rate of fuel = 1.05 Kg/min

The above computed data will help us to design the fuel supply system

Let
A/F ratio = 32:1

Reasons:

1. The Operating Range of A/F ratio for C.I engines is (18 to 70)
2. The Stoichiometric A/F ratio for Diesel fuel = 14.5
3. The diesel engines are operated lean to minimize the soot emissions from diesel engines. However this does not decrease the power output from the engine.
4. The use of a heavy duty compressor for the design of a heavy duty truck engine results in a much larger mass of air entering the engine cylinders during intake process.
5. Therefore in order to develop a large power from a heavy duty truck engine, the mass of fuel entering the engine cylinders is also much higher.
6. The higher mass of air and fuel in the cylinders of a heavy duty truck engine increases the soot emissions further. This is controlled by making the air to fuel ratio further leaner for the design of heavy duty truck engines.

A/F = $\frac{\dot{m}_a}{\dot{m}_f}$

Where

$\dot{m}_a$ = mass flow rate of air
$\dot{m}_f$ = mass flow rate of fuel

From the above equation:

$\dot{m}_a$ = A/F * $\dot{m}_f$
$\dot{m}_a$ = 32 * 1.05 = 33.6 Kg/min

Mass flow rate of Air = 33.6 Kg/min
The Above computed data will help us in the design of air supply system
This includes the inclusion of the heavy duty compressor of a heavy duty turbocharger

Volumetric Efficiency is given by the equation:

ηv = $\frac{2*\dot{m}_a}{\rho_{a,i}V_d N}$

Where

$\dot{m}_a$ = Actual mass flow rate of air
$\dot{m}_a$ = 33.6 Kg/min
$\rho_{a,i}$ = ambient inlet air density
$\rho_{a,i}$ = 1.18 Kg/m^3
$\rho_{a,p}$ = density of air after compression by a heavy duty compressor at intake port of the engine.
$\rho_{a,p}$ = 3.3 Kg/m^3
Volumetric efficiency of the diesel engine without including a compressor = 0.91 = 91%
Volumetric efficiency of a diesel based heavy duty truck engine after including a heavy duty compressor = 2.79 = 279%
V_d = Engine displacement volume ---- ?

Substituting the values in the equation for volumetric efficiency we get:

V_d = 0.01243 m^3

V_d = 12430 cc

V_d = 12400 cc
Displacement Volume Required = 12.4 liters

Conclusions:

To design a Turbo-charged heavy duty truck engine for the rated power of 300 KW at the suitable rated speed of 1800 rpm with a much leaner combustion for the control of soot emissions
V_d = 12400 cc

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Text Book:
Vehicular Engine Design
By Kevin L. Hoag
Published By: SAE International USA