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SIMULATION OF THE AIRFLOW CHARACTERISTICS OF A TWO-STROKE NATURAL GAS ENGINE WITH AN ARTICULATED CRANK

J. Adair
Mechanical Engineering Department
Colorado State University
Fort Collins, CO 80523

A. Kirkpatrick
Mechanical Engineering Department
Colorado State University
Fort Collins, CO 80523

D. B. Olsen
Mechanical Engineering Department
Colorado State University
Fort Collins, CO 80523

H. Gitano-Briggs
Mechanical Engineering Department
Colorado State University
Fort Collins, CO 80523

ABSTRACT

The topic of this paper is the simulation of the airflow characteristics of a large bore two stroke natural gas fueled engine. The modeling was performed with the program WAVE, a computer code developed for engine cycle simulations. The engine studied was a four cylinder Cooper GMV engine. This engine has an articulated crankshaft connecting even and odd bank cylinders. Due to the articulation, the even bank cylinders have different piston profiles, port profiles, and compression ratios than the odd bank cylinders.

Due to the non-symmetric timing and articulated geometry of the odd and even banks, the gas flow processes are not the same for each cylinder bank. The different manifold and port pressure profiles result in different amounts of trapped mass in the odd and even banks. The even bank is predicted to have a smaller amount of trapped mass and slightly lower trapping and scavenging efficiencies. Finally, the model predicts that the even bank cylinders attain higher maximum temperatures, which would produce increased NOx.

INTRODUCTION

The topic of this paper is the simulation of the airflow characteristics of a large bore two stroke natural gas fueled engine. Large bore (bore >30 cm.) slow speed (speed <500 rpm) two stroke engines are used in stationary applications such as gas compression and electric power generation. There are about 8,000 engines of this type used in the United States. The specific engine modeled in this paper, the GMV Cooper Bessemer engine, is widely used in the gas compression industry, primarily in 10 cylinder versions. This engine has some unique characteristics, including an articulated crankshaft and a V- bank configuration. A schematic of the engine is shown in Figure 1.

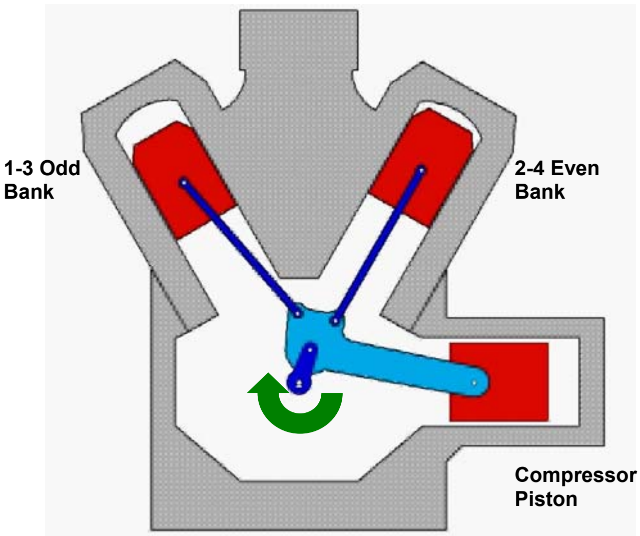

Fig. 1. Articulated crank geometry of Cooper-Bessemer Engine

The articulated geometry results in significant differences in piston motion between the two sides of such an engine. The crank throw pins for the 4 power pistons are offset from the main crank throw pin for the compressor piston. As a consequence, the crank throw pins for the power pistons prescribe different elliptical orbits. The port timing for the odd and even bank is given in Table 1. The differences in motion

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GMV-4TF Port Geometry		
	1-3 Odd Bank	2-4 Even Bank
Top Dead Center	0	0
Bottom Dead Center	188.0	181.5
Exhaust Port Open	116.0	114.5
Intake Port Open	135.0	131.5
Intake Port Close	-123.5	-129.0
Exhaust Port Close	-107.0	-112.5
Exhaust Port Open Duration	137.0	133.0
Intake Port Open Duration	101.5	99.5

Table 1. Porting differences between the odd and even banks of the GMV Engine.

between the left and right engine banks result in the odd bank exhaust ports being open for about four additional degrees and the intake ports being open for two additional degrees relative to the even bank.

A consequence of the V-bank and the articulated crankshaft geometry is that the four cylinder GMV is an odd-fire engine, meaning that the timing between TDC of all cylinders is not equal. The timing between TDC from cylinder 1 to cylinder 2 is 62.5 degrees, from 2 to 3 is 117.5 degrees, and from 3 to 4 is 62.5 degrees, giving the engine a dynamic imbalance. This issue is negated in engines with cylinder numbers in multiples of 6, and reduced in other engines with cylinder counts more that 6. The scavenging in this engine is classified as Schnurle or loop scavenging, in which the intake ports are angled upward to reduce short circuiting, as shown in Figure 2. The scavenging flow field in this cylinder is discussed further in Boyer et al. [1].

ENGINE SIMULATION

The engine simulation was performed using the engine simulation program WAVE [2]. In this program, the components of the engine are represented as either a duct or a junction. A cylinder is a special type of junction in WAVE. Engine characteristics such as combustion, scavenging, engine geometry and heat transfer, are defined in the cylinder junction. The equations for conservation of mass and energy are applied to each duct or junction. The conservation of momentum equation is applied to the boundary between each volume. The four cylinder model is shown in Figure 3. The separate piston motions were modeled using the parent – child feature.

Operating parameters used for the engine model are provided in Table 2. An instrumented four cylinder Cooper GMV engine at the CSU Engines and Energy Conversion Laboratory was used to provide cylinder pressure and heat release test data. The inlet manifold pressure is boosted above atmospheric pressure by a separately powered supercharger. The natural gas fuel is directly injected into the cylinder, so that the scavenging gas is air. The engine is operated at 300 rpm. Further information about the engine and associated test data is given in Olsen et al. [3].

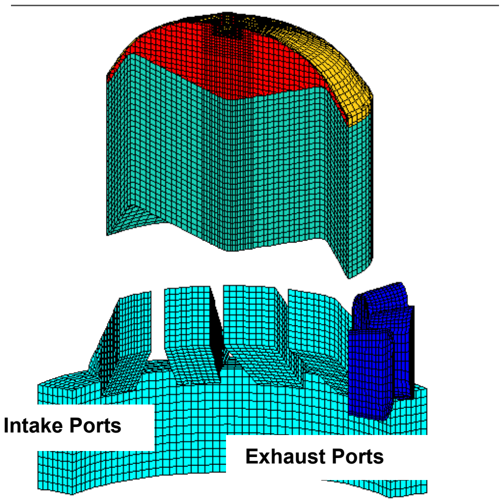

Fig. 2 Cross section of intake and exhaust ports.

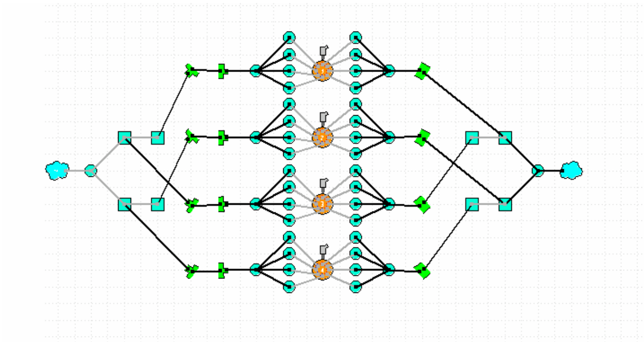

Fig. 3 WAVE schematic of GMV Engine.

Input Parameter	Odd Bank	Even Bank
Pressure:		
Inlet	25.4 kPa	25.4 kPa
Exhaust	17.0 kPa	17.0 kPa
Fluid Temperature:		
Inlet	316.5 K	316.5 K
Exhaust	600 K	600 K
Scavenging Model	Fully Mixed	Fully Mixed
Heat Transfer Multiplier:		
Piston Top	1.23	1.23
Cylinder Head	1.36	1.36
Friction Multiplier	3.31	3.31
Wiebe Function:		
50% Burn Location	13.8 deg	12.7 deg
10%-90% Burn Duration	22 deg	22 deg
Exponent	2.676	2.676
Stroke	35.6 cm	35.5 cm
Compression Ratio	12.08	11.75

Table 2 Input parameters for the WAVE simulation

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The discharge coefficient profiles for the intake and exhaust ports were obtained from Heywood and Sher [4] as indicated in Figures 4 and 5. The profile in Figure 4 is for intake ports inclined at 48 degrees, close to the 51-degree inclination on the GMV engine. The exhaust ports are not inclined. The profile for the exhaust ports was limited to being a function of port open fraction only, and did not include the pressure ratio dependence. The computed overall air flow per cylinder through the engine was matched to the overall airflow (0.233 kg/s/cylinder) in the actual engine by modifying the intake and exhaust port flow coefficients by a common multiplier of 1.06.

When the actual engine is operated, the mass of fuel injected into each cylinder is adjusted to obtain a maximum

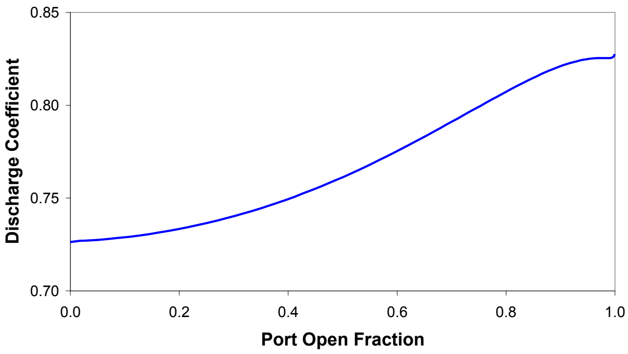

Fig. 4 Intake port discharge coefficient profile.

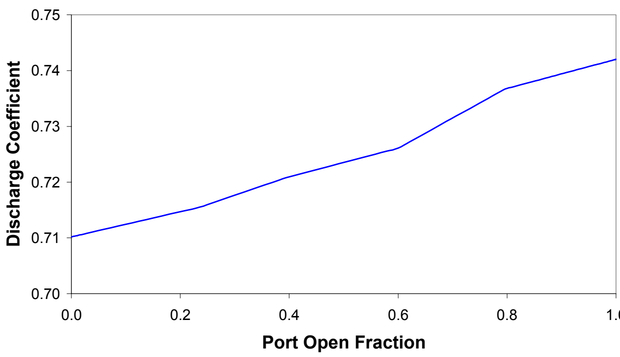

Fig. 5 Discharge port coefficient profile.

cylinder combustion pressure of 34.5 bar (500 psi), as shown in Figure 6a. The ignition timing is also adjusted so that the peak pressure in each cylinder occurred at 18 degrees after TDC. Matching of peak pressure and location of peak pressure for each bank was accomplished in the model in a similar fashion by adjusting the fuel delivered to each cylinder and adjusting the 50% burn location of the Wiebe function, as indicated in Table 2. The Wiebe function exponent and 10-90% burn duration required for the heat release modeling were obtained from averaged experimental data for each cylinder bank.

SIMULATION RESULTS

The overall simulation results are summarized in Table 3, Note that the even bank required a greater amount of fuel to balance all of the cylinder maximum pressures at 34.5 bar.

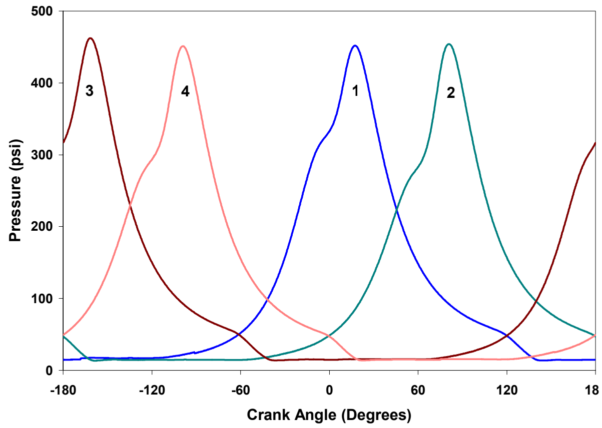

Fig. 6a Measured cylinder pressure profiles for all four cylinders.

The even bank is predicted to have a about a 10% lower airflow rate than the odd bank.

Various airflow parameters, the trapping efficiency, the trapped air/fuel ratio, the delivery ratio, and the scavenging efficiency are tabulated and compared with experimental data in Table 4. The predicted airflow parameters are almost the same for both sides, with the exception of the trapped air/fuel ratio, which is about 10% higher on the odd bank.

The trapping efficiency is defined as the mass of delivered air retained divided by the mass of delivered air. The trapped air/fuel ratio is the air/fuel ratio multiplied by the trapping efficiency. The scavenging efficiency is defined as the mass of delivered air retained divided by the mass of trapped cylinder charge. The delivery ratio is the mass of delivered air per cycle divided by the displaced volume and ambient density.

	Odd Bank	Even Bank	Average
Fuel Mass (mg)	1055	1080	1068
Air Flow (kg/s)	0.244	0.222	0.233
Brake Power (kW)	82.4	82.7	82.5
Max. Pressure (bar)	34.6	34.8	34.7
IMEP (bar)	5.43	5.45	5.44

Table 3. Overall engine performance results.

		Odd Bank	Even Bank
Trapping Efficiency	Simulation	0.49	0.48
	Experiment	0.54	
Trapped A/F Ratio	Simulation	22.5	19.8
	Experiment	21.3	
Delivery Ratio	Simulation	1.77	1.77
Scavenging Efficiency	Simulation	0.86	0.85

Table 4. Engine airflow characteristics.

The overall measured trapping efficiency was measured directly using the tracer gas method as described in Olsen et al. [3]. The computed trapping efficiencies of about 0.49 were lower than the overall measured value of 0.54. One possible

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explanation for this discrepancy is modeling of the port flow with the flow coefficient multiplier, which was the same for both the intake and exhaust ports.

Figure 6b shows the computed pressure traces for all four cylinders of the engine. The plot shows the cylinder pressure at any time during the cycle as a function of cylinder #1 crank angle degrees. The maximum pressure for all four cylinders is approximately 34.5 bar (500 psi). The maximum pressure for each cylinder takes place at 18 degrees after relative TDC. The pressure traces produced by the model are very similar to those produced by the engine as seen in Figure 6a. The odd-firing sequence is evident, with cylinders 1 and 2 separated by 62.5 degrees, similarly for cylinders 3 and 4.

Figure 7 shows the cylinder trapped mass at any time during the cycle relative to cylinder #1 crank angle degrees. Cylinder #1, for example, begins blow-down at approximately +120°, when the exhaust port is uncovered. At this point the amount of mass in the cylinder rapidly decreases. Once blow-down is complete at about +140° the relatively dense scavenging air begins to enter the cylinder. As it mixes and displaces the hot combustion products, the amount of mass in cylinder #1 increases. The mass in the cylinder increases to a maximum value and at about -140° begins to decrease. The decrease in the cylinder mass from about -140° to -120° represents the time in the cycle when the intake ports are closed, but the exhaust ports are open, and the piston is moving towards the head. Cylinder gas is forced out of the cylinder during this period. At about -120° the exhaust ports are effectively closed, and the slight increase in mass that follows is from fuel injection. At roughly -90° the cylinder #1 fuel injection is complete, and the amount of mass in the cylinder is fixed.

Figure 7 illustrates the cylinder trapped mass differences between the two banks of the engine. The amount of mass in the cylinders of the odd bank is higher than the amount of mass in the even bank cylinders. The minimum mass in each cylinder, however, is about the same. Consequently, the odd bank has a larger change in trapped mass than the even bank and therefore has a slightly higher scavenging efficiency.

The temperature profiles, seen in Figure 8, were analyzed using a two-zone combustion model. The higher line for each cylinder represents the temperature for the burned gases. The lower line for each cylinder represents the temperature of the unburned gases. The profile for the unburned gases appears identical for all four cylinders of the engine. The odd bank has a relatively lower burned gas maximum temperature. One explanation is that the greater amount of trapped mass in the odd bank increases the total specific heat of the cylinder mixture, which reduces the temperature change during combustion.

NOx formation is exponentially dependent on temperature. Thus, the lower burned gas temperature of the odd bank cylinders results in lower NOx formation. The even bank attains a higher maximum temperature. Because the even bank remains at an elevated temperature for approximately 40 crank angle degrees, the even bank is predicted to produce more NOx than the odd bank. This behavior is consistent with engine test results. Test results indicate that the even bank produces about twice as much NO_X and 15% more formaldehyde than the odd bank.

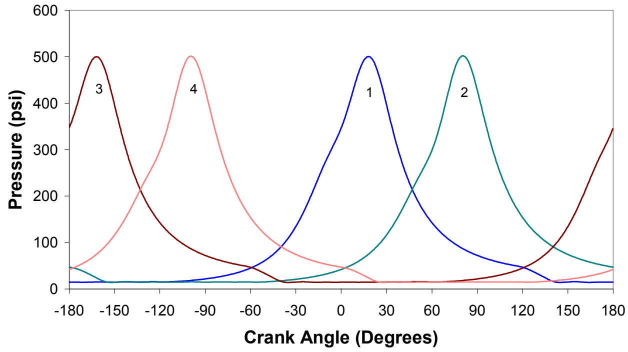

Fig. 6b Computed cylinder pressure profiles for all four cylinders.

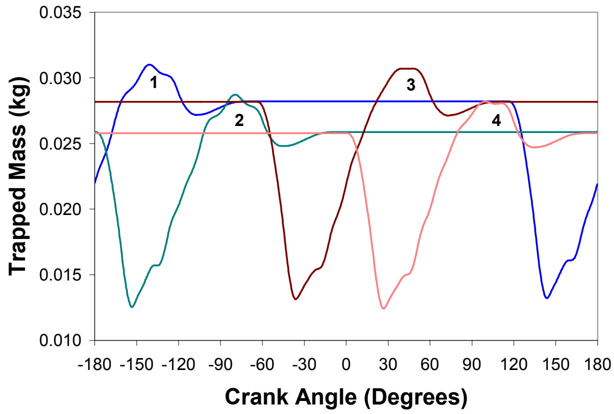

Fig. 7 Trapped mass profiles for all four cylinders

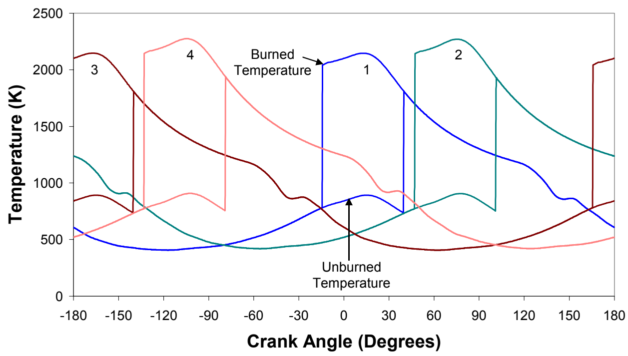

Fig. 8 Burned and unburned gas temperature profiles.

As seen in Figure 9, the even bank has a higher heat transfer rate. This higher heat transfer rate is due to the temperature difference of the two banks. Because the even bank attains a higher maximum temperature, it also attains a higher maximum heat transfer rate. The fact that the odd bank has a larger amount of trapped mass does not appear to have any effects on the heat transfer rate; and temperature is the dominant effect. Also note the sharp increase in heat transfer rate during blow-down for each cylinder. As the ports open the heat transfer in the cylinder increases due to the increase in the velocity of the cylinder gas.

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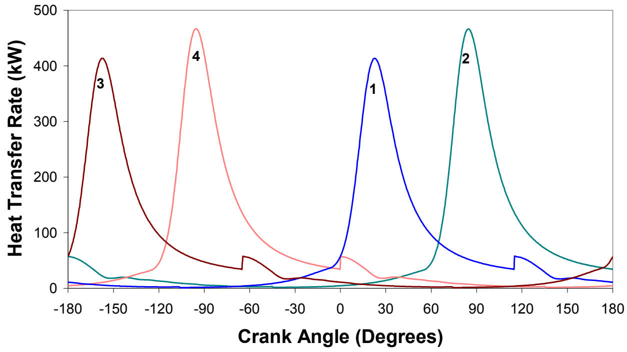

Fig. 9 Cylinder heat transfer profiles.

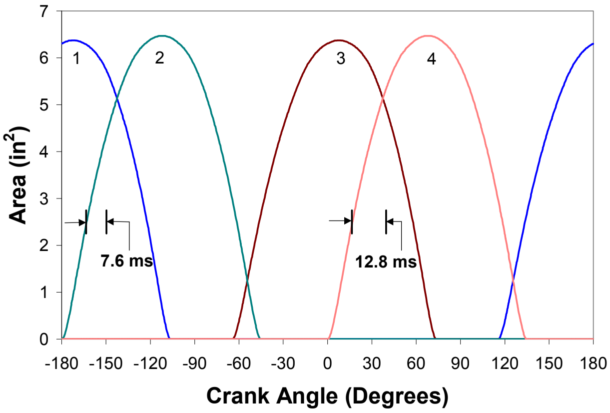

Fig. 10 Exhaust port area profiles versus crank angle (relative to Cylinder #1 TDC).

The articulated crank geometry also has an effect on the exhaust manifold pressure wave behavior. The exhaust port timing profile is shown in Figure 10. As the exhaust ports for cylinder #2 are opening, the exhaust ports for cylinder #1 are closing. Likewise, as the exhaust ports for cylinder #4 are opening, the exhaust ports for cylinder #3 are closing. As the exhaust ports open, a pressure wave from blow-down travels through the exhaust manifold via the cylinder exhaust elbows and into the main exhaust elbow where exhaust from the two sides of the engine combine. The exhaust manifold is separated between the banks. The pressure wave from the blow-down then travels back down the opposite side of the exhaust manifold where it reaches the exhaust ports of an opposite bank cylinder. Therefore, the blow-down pulses from the even cylinders act as plugging pulses and reduce the charging air loss from the odd cylinders, which explains why the predicted trapped mass is greater in the odd cylinder bank.

The wave dynamics in the exhaust are extremely complex, as discussed by Boretti et al. [4] and Blair [5]. The predicted paths of the pressure waves from the blow-down of cylinders 2 and 4 are illustrated in Figure 11. The timing of the engine allows the pulses from the even bank to arrive at the odd bank immediately before the ports close, but not vice versa. For example, the pressure wave from cylinder #2 travels through the exhaust system and arrives at cylinder #1 in about 7.6 ms (~15^{o} CA) while its exhaust ports are still open. Likewise, the pressure wave from cylinder #4, travels through the exhaust system and arrives at cylinder #1 in about 12.8 ms (~25^{o} CA).

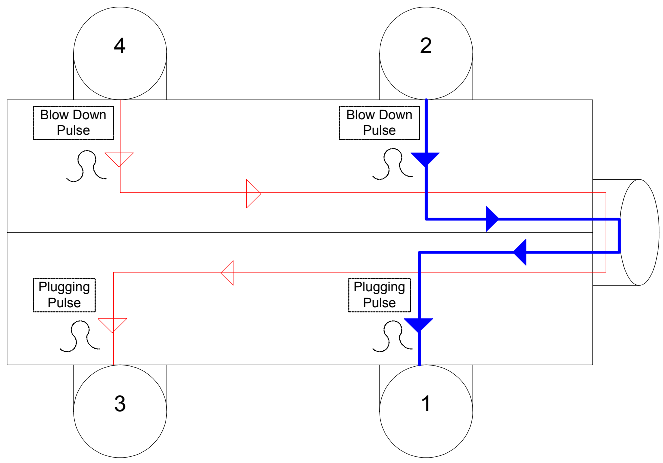

Fig. 11 Schematic of exhaust manifold pressure waves.

SUMMARY AND CONCLUSIONS

The airflow characteristics of a two-stroke natural gas engine with an articulated crank were determined using an engine simulation program. Due to the non-symmetric timing and articulated geometry of the odd and even banks, the gas flow processes are not the same for each cylinder bank. The different manifold and port pressure profiles result in different amounts of trapped mass in the odd and even banks. The even bank is predicted to have a smaller amount of trapped mass and slightly lower trapping and scavenging efficiencies. Finally, the model predicts that the even bank cylinders attain higher maximum temperatures, which would produce increased NOx.

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