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*Published in:*
SAE Technical Paper Series

1998

*SAE TECHNICAL PAPER SERIES*

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**International Fall Fuels and Lubricants Meeting and Exposition**
**San Francisco, California**
**October 19-22, 1998**

**400 Commonwealth Drive, Warrendale, PA 15096-0001 U.S.A. Tel: (724) 776-4841 Fax: (724) 776-5760**

**ISSN 0148-7191**


**Printed in USA**

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## ABSTRACT

This paper is a direct continuation of a previous study that addressed the performance and design of a variable compression engine, the Alvar-Cycle Engine [1]. The earlier study was presented at the SAE International Conference and Exposition in Detroit during February 23-26, 1998 as SAE paper 981027. In the present paper test results from a single cylinder prototype are reviewed and compared with a similar conventional engine. Efficiency and emissions are shown as function of speed, load, and compression ratio. The influence of residual gas on knock characteristics is shown. The potential for high power density through heavy supercharging is analyzed.

## INTRODUCTION

VARIABLE COMPRESSION RATIO – To achieve variable compression ratio, an engine design where it is possible to change the nominal or effective compression ratio is needed. The purpose is to:

- Increase the fuel efficiency at part load by increasing the compression ratio.
- Increase power density through high boost pressure (turbo or mechanical supercharger) by reducing the compression ratio to avoid knock.
- Optimize the efficiency and/or emissions depending on the fuel octane number by varying the compression ratio and ignition timing.

Because of the reasons mentioned it is an advantage if the compression ratio could be changed continuously when the load or the fuel is changed (Flexible Fuel Vehicle, FFV). Nominal and effective compression ratio are defined as:

$$r_{c,nominal} = \frac{V_{max}}{V_{min}} = \frac{V_d + V_c}{V_c}$$

$$r_{c,effective} = \frac{V_{IVC}}{V_{min}} \qquad \text{(Eq. 1)}$$

Several concepts with variable compression ratio (VCR) have been proposed, for example designs where the cylinder and cylinder head is moved up or down from the crankshaft or designs with volume controlling pistons in the combustion chamber.

## THE ALVAR - CYCLE ENGINE

The Alvar engine is a patented 4-stroke engine that provides variable compression ratio. The engine has derived its name from the inventor, Mr. Alvar Gustavsson from Skärblacka, Sweden. The engine can be either spark or compression ignited. The spark ignition case is evaluated in this paper.

THE ALVAR ENGINE PRINCIPLE – The Alvar engine uses a conventional engine block with a modified cylinder head as shown in Figure 1. The cylinder head contains besides camshaft, valves and sparkplugs of conventional type, smaller extra cylinders with pistons, conrods and a crankshaft. These cylinders share the same combustion chamber as the main cylinders. The crankshaft for the extra cylinders, called secondary shaft, is connected to the engine main crankshaft via some kind of a phaseshift mechanism and a transmission that gives the secondary shaft a speed of half that of the main crankshaft speed. The transmission for the secondary shaft can thereby be combined with the camshaft transmission.

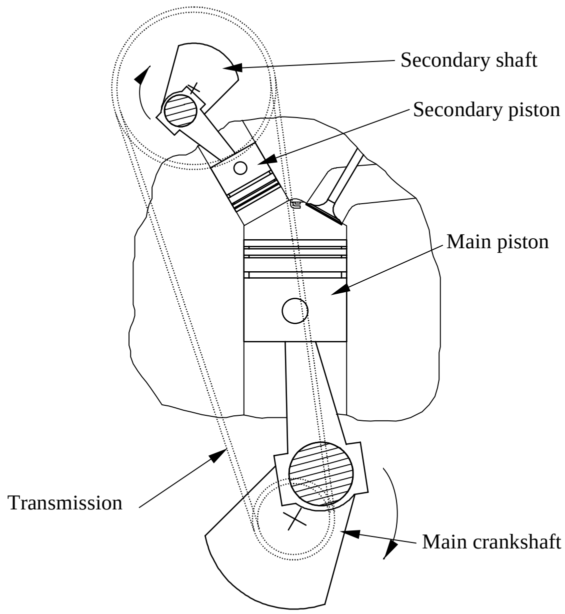


Figure 1. Parts in the Alvar engine. The figure shows the beginning of the expansion stroke.

PHASE SHIFT – The secondary shaft is always phase-shifted an angle i.e. when the main piston is at TDC after compression, the secondary piston has already passed TDC and the angle from the secondary TDC position to the actual position measured on the secondary shaft is called phaseshift. shows the definition of phaseshift. By changing this phaseshift different compression ratios are achieved. Higher phaseshift gives lower compression ratio and vice versa.

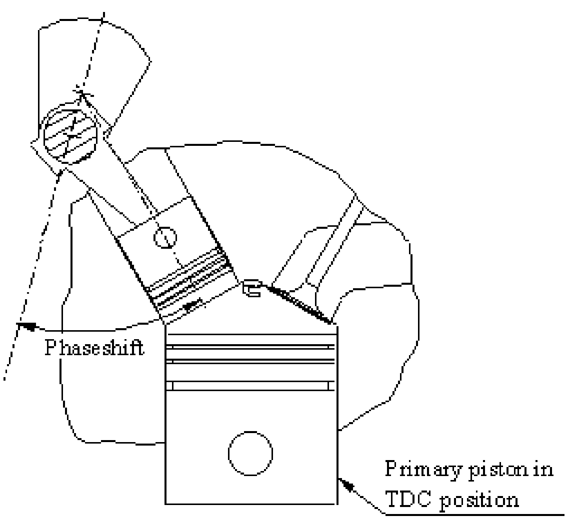


Figure 2. Definition of phaseshift. The primary piston is in its TDC position at end of compression. Phaseshift is the angle on the secondary shaft between the position when the secondary piston is at its "TDC" and the actual position.

A commercial Alvar engine requires some sort of phase-shift mechanism. What design is appropriate for this application is not yet clear. Test results for a simple mechanical design were reviewed in [1].

SECONDARY SHAFT TORQUE – Under part load conditions the engine is throttled and during the intake stroke, Figure 3, the secondary piston is moving towards TDC and the torque on the secondary shaft will thereby be positive. During the compression stroke, Figure 4, and at least the early part of the expansion stroke, the secondary piston is moving away from TDC, and the torque on the secondary crankshaft will thereby be positive again. Totally, the torque will always be positive during the whole cycle. Results indicates that there is a linear relationship between the torque on the secondary shaft and the torque on the main shaft [1]. This torque gives a power flow out from the secondary shaft that is transferred via the transmission and the phase shift mechanism to the main crankshaft.

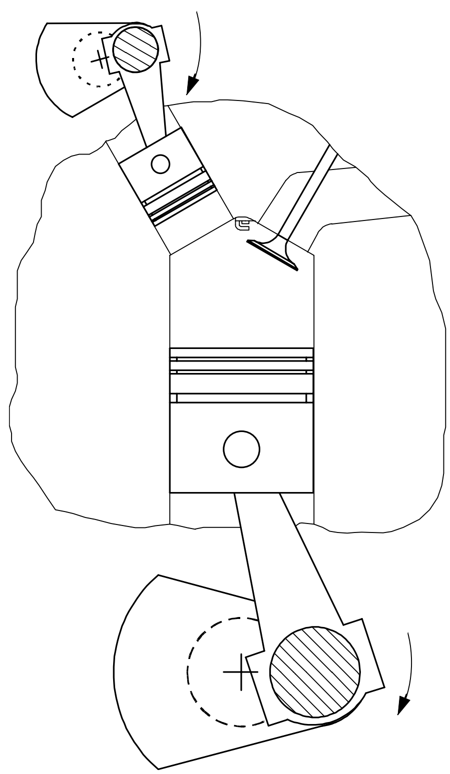

Figure 3. Intake stroke, phaseshift = 60°

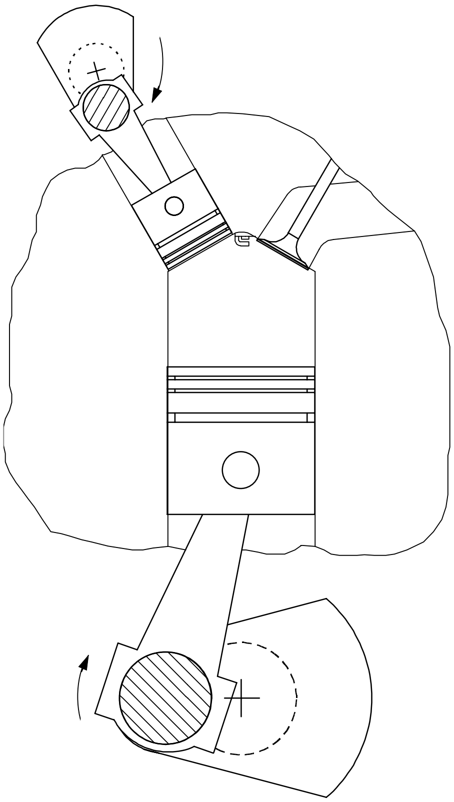

Figure 4. Compression stroke, phaseshift = 60°

RESIDUAL GAS FRACTION – When high compression ratio is set, as shown in Figure 5, the minimum volume during intake stroke is high because the secondary shaft rotates with half of the speed of the main shaft. Thereby the level of internal residual gas (gas from previous cycle) will be high. Higher levels of residual gas gives lower NOx-emissions and a higher knock limit since residual gas has a cooling effect. At low compression ratio, Figure 6, the minimum volume becomes low and the residual gas level is thereby low. In a way this is a built-in EGR control in the Alvar engine. The amount of residual gas depends on the secondary cylinder stroke and bore.

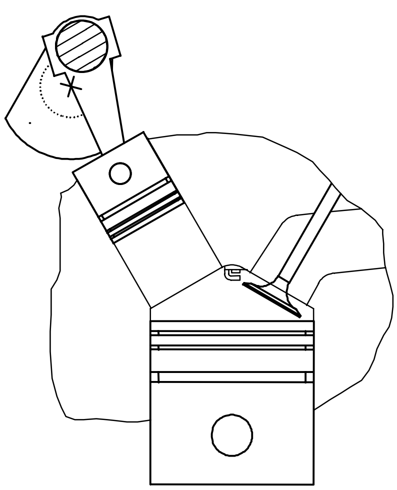

Figure 5. End of exhaust stroke. High compression ratio gives high amount of residual gas during gas exchange.

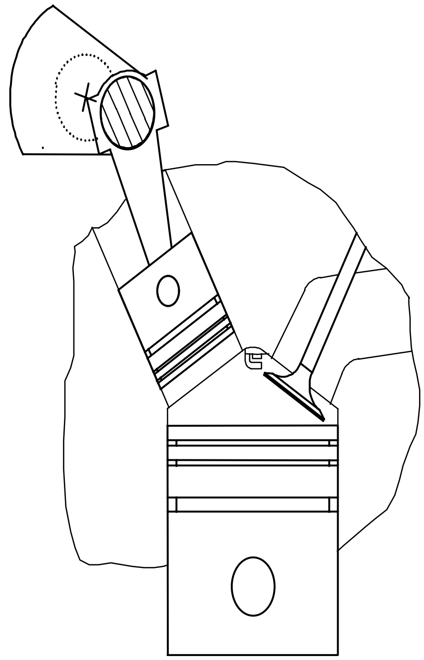

Figure 6. End of exhaust stroke. Low compression ratio gives low amount of residual gas during gas exchange.

## OBJECTIVE

The purpose of this test was to investigate if the Alvar engine gives higher efficiency at lower loads and higher power density through supercharging. Another purpose was to examine the emission characteristics of this engine. It was suspected to give significantly high hydrocarbons emissions due to the increase in total piston circumference, but it did not.

TEST PROGRAM – The program for the tests is described below and the principle is shown in Figure 7.

- Fuel: Isooctane, all tests at l=1.
- Speed: 2000 and 3000 rpm.
- Compression ratios: 15, 13 10 and 8 (approx.).

For each speed and compression ratio (the numbers refer to Figure 7) :

1. MBT - spark advance ($ADV_{MBT}$) was determined by tests at part load and spark advance between 0° and 45°. Knock was avoided. For each advance cylinder pressure, torque, emissions, fuel flow and various temperatures and pressures were measured.
2. At $ADV_{MBT}$; the inlet air pressure was increased in steps of 0.1 bar until knock occurred. The same measurements and as in point 1 were done for each inlet pressure.
3. The ADV was reduced in steps of 5° and inlet air pressure was increased in steps of 0.1 bar until the knock limit was known. The same measurements and calculations as in point 1 were done. This was repeated until ADV=0° or the $T_{exhaust}$=980°C limit was reached.

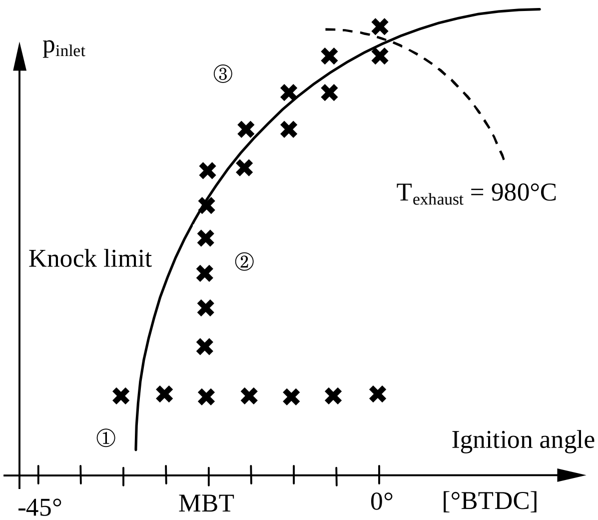


Figure 7. Test program. This is not an actual case but shows the principle.

Optimum spark advance and maximum IMEP for each speed, compression ratio and inlet air pressure were determined. The same procedure was used with a conventional cylinder head that was modified for different compression ratios. When the engines were supercharged an external compressor was used. This was not compensated for when the efficiencies were calculated.

## EXPERIMENTAL APPARATUS

ENGINE SPECIFICATION – The engine that was used in these tests is based on a Volvo B5254 FS (Volvo 850 car engine), 2435 $cm^2$ displaced volume. One of the five cylinders was used since manufacturing a modified cylinder head and crankshaft for five cylinders would be a too expensive solution for this project. The Alvar engine was parametrically studied at MIT [2] and the manufacturing of the cylinder head was made at Adiabatics Inc. The engine specification is shown in Table 1 and the compression ratios that were tested are shown in Table 2. Nominal compression ratio (NCR) means geometrical compression ratio. Effective compression ratio (ECR) means when inlet valve closing is considered. Cylinder volumes and residual gas fraction at this compression ratios are shown in Table 3. The combustion chamber volume was measured and the volumes and compression ratios were then calculated from this measurement. The engine was equipped with two spark plugs. Ignition and fuel injection were controlled by a separate computer. Special pistons were designed to be able to reach higher compression ratios. In Figure 8 cylinder heads and pistons are shown. The piston for the Alvar engine has two slots for the valves in case the camshaft transmission, that is combined with the secondary shaft transmission, would malfunction.

Table 1. Engine specifications. Primary geometry specifications and valve timing are equal to the standard engine.

| Geometry | [mm] |
|---|---|
| Primary | |
| Bore | 83,0 |
| Stroke | 90,0 |
| Offset | 0,8 |
| Secondary | |
| Bore | 40,0 |
| Stroke | 40,9 |
| Offset | 10,0 |

| Valve timing | [°] |
|---|---|
| Inlet open | 8 BTDC |
| Inlet close | 52 ABDC |
| Exhaust open | 44 BBDC |
| Exhaust close | 16 ATDC |

Table 2. Calculated compression and expansion ratios at different phaseshifts.

| Phase-shift | Compression ratios | | Expansion ratios | |
|---|---|---|---|---|
| | Nominal | Effective | Nominal | Effective |
| φ [°] | NCR | ECR | NER | EER |
| 0,00 | 15,03 | 12,96 | 15,00 | 13,34 |
| 34,29 | 12,79 | 11,04 | 13,45 | 12,05 |
| 60,00 | 10,29 | 8,96 | 11,14 | 10,05 |
| 85,71 | 8,36 | 7,37 | 9,18 | 8,33 |
| 137,10 | 6,51 | 5,86 | 6,95 | 6,39 |

Table 3. Calculated cylinder volumes during compression stroke and calculated residual gas fraction = {minimum volume during exhaust stroke}/$V_{max}$

| Phase-shift | Compression ratios | | Expansion ratios | |
|---|---|---|---|---|
| | Nominal | Effective | Nominal | Effective |
| φ [°] | NCR | ECR | NER | EER |
| 0,00 | 15,03 | 12,96 | 15,00 | 13,34 |
| 34,29 | 12,79 | 11,04 | 13,45 | 12,05 |
| 60,00 | 10,29 | 8,96 | 11,14 | 10,05 |
| 85,71 | 8,36 | 7,37 | 9,18 | 8,33 |
| 137,10 | 6,51 | 5,86 | 6,95 | 6,39 |

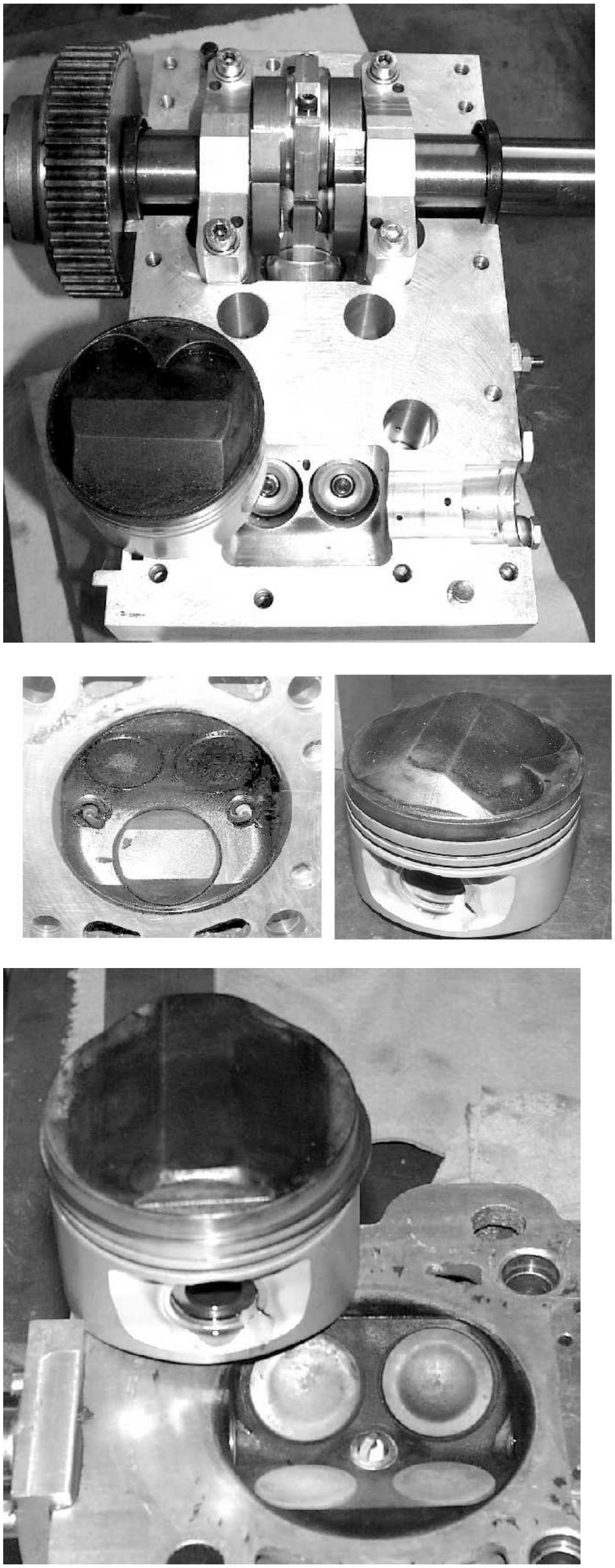

Figure 8. Top: Alvar engine cylinder head without the cover and camshaft mounted. Middle: Alvar engine combustion chamber and piston. Bottom: Standard engine combustion chamber and high compression piston.

MEASUREMENT SYSTEM – The cylinder pressure was measured with a Kistler 6043-A60 piezo-electric transducer. The signal was amplified with a Kistler 5011 charge amplifier, transformed in a 100kHz A/D card and stored on a PC. 200 cycles from each point were stored with a resolution of 0.2 degrees. A HP-logger connected to a PC was used to measure arbitrary temperatures and pressures. Measurements were done over periods of 3 minutes. CO (NDIR), $CO_2$ (NDIR), $O_2$ and HC (FID) emissions were measured with Cussons exhaust gas analyzers. NOx were measured with a TECAN chemiluminescence analyzer.

EVALUATION – The pressure data were evaluated through a one-zone, heat release analysis using Woschni´s heat transfer model. This analysis was done in a heat release program modified for the Alvar engine geometry. More information and a detailed description on heat release analysis can be found in [3]. The results from these calculations were further processed together with emission data and fuel flow in MATLAB.

## RESULTS

The results are compared with data from the same base engine with a standard Volvo cylinder head, also one cylinder. The Alvar engine used one inlet and one exhaust valve from the four-valve-per-cylinder standard engine. This standard cylinder head was also tested at different compression ratios. The compression ratio was changed by using a special piston and by using one or two cylinder head gaskets. The expressions ALV and STD in the figures mean Alvar engine and standard engine respectively. The number that follows is the approximated nominal compression ratio. See Table 2 for more exact compression ratio information. Only two speeds were tested, 2000 and 3000 rpm due to time limitations. In each of the following figures, results from both these speeds are presented. The results that are presented here are without any knock present i.e. points strictly below the knock limit.

SPEED, LOAD AND SPARK ADVANCE – Figure 9 shows the inlet pressures and ignition timing that were used. Inlet pressure is reported as absolute pressure. Compare this figure with the initial test program in Figure 7. Since the test program is in a grid pattern the results were affected by this, some of the curves that follow are slightly zigzag shaped. At 2000 rpm, for the Alvar engine at compression ratio 10 the tooth belt for the secondary shaft slipped a few teeth by itself and this was not discovered until the heat release analysis was done. Therefore only two points are shown from this compression ratio.

INDICATED MEAN EFFECTIVE PRESSURE – Net and gross indicated mean effective pressure are shown in Figure 10 and Figure 11 respectively. Gross means without exhaust and intake stroke i.e. without pump or throttle losses. Both in the Alvar and the standard engine cases, we were able to reach almost 2 bar absolute in inlet pressure at lower compression ratios and this resulted in almost 20 bar IMEP-net. The load increased more or less linearly with the inlet pressure as expected. The standard engine gave higher IMEP than the Alvar engine at a specific inlet pressure. At 2000 rpm the difference was small, especially at the highest compression ratio, but at 3000 rpm it shows clearly that there are large losses over the valves. This is not surprising since the Alvar engine only has half the total valve area compared to the standard Volvo engine. The results show that through variable compression ratio it is possible to increase the maximum output by lowering the compression ratio. At higher compression ratios and high load, the spark advance had to be reduced extensively and this caused low efficiency and high exhaust gas temperatures.

INDICATED EFFICIENCY – Net and gross indicated efficiencies are shown in Figure 12 and Figure 13, and again, note that in the STD 10 case, the mixture was richer than stoichiometric. The figures show clearly how the efficiency first increases up to a maximum and then decreases rapidly, especially at high compression ratios. The reason for this drop in efficiency is that the spark advance has to be reduced to avoid knock when the load increases. The MBT timing is thereby abandoned and the efficiency drops. Higher compression ratio requires larger reduction in spark advance. At low loads there is a clear increase in efficiency with higher compression ratios as expected. The Alvar engine has slightly higher efficiency at part load and high compression ratio than the standard engine. It is possible that this depends on the higher amount of residual gas in the Alvar engine case. Higher amounts of residual gas give lower losses over the throttle at a specific (part-) load and it dilutes the fuel air mixture. These two effects both result in higher efficiency. Note that an external compressor was used under supercharged conditions and this is not compensated for. The results clearly show that through variable compression ratio, higher efficiencies at part load conditions can be achieved.

EMISSIONS – All emissions were measured dry except NOx that was calculated to dry emissions.

<u>CO emissions</u> – The CO-emissions are shown in Figure 14. Note that in the standard engine STD 10 case the CO emissions was very high, this is because we had some trouble with the equipment for the emission measurements and this caused a too rich mixture. This was confirmed by the low combustion efficiency and the high HC emissions in this case. The other cases are below 30 g/kWh in CO and it is difficult to see differences between the standard and the Alvar engine. Since the mixture was set manually, although with aid from EGO - sensor and emission information, there were small deviations and the mixture sometimes became slightly too rich or lean and this had a large effect on CO emissions.

<u>HC emissions</u> – HC emissions, propane equivalent, are shown in Figure 15. The HC emissions were expected to be higher in the Alvar engine case. At 3000 rpm there is a clear but small difference. The Alvar engine had slightly higher HC emissions. The HC emissions were almost exactly the same at 2000 and 3000 rpm for the Alvar engine. At very low and high loads the HC-emissions increased both for the Alvar engine and the standard engine but this is not surprising since the pressures and temperatures were lower during combustion in these cases and this reduces the combustion efficiency.

<u>NOx emissions</u> – NOx emissions are shown in Figure 16 and Figure 17. The NOx emissions were lower at higher compression ratios but this can be explained that for a high compression ratio the spark advance had to be reduced more to avoid knock than in the low compression ratios. Thereby the pressures, temperatures and the time that the gas is exposed to this temperatures were reduced and this probably limited the NOx formation. It is difficult to see a difference between the Alvar and the standard engine and thereby any residual gas effect on NOx formation can not be seen.

COMBUSTION EFFICIENCY – The combustion efficiency shown in Figure 18 is calculated from emissions according to [4]. The average combustion efficiency is approx. 95% for both engines.

VALVE FLOW LOSSES – The IMEP results in Figure 10 and Figure 11 indicated high losses over the valves. This is confirmed by Figure 19 that shows the p – V diagram over the intake and exhaust stroke at WOT conditions. At the intake stroke there is a large pressure drop over the intake valve in the Alvar engine case. This also limits the amount of gas that will be expanded and pushed out during exhaust stroke. The IMEP is of course higher in the standard engine case since more gas is induced and this can also explain why the pressure over the exhaust valves in the standard engine is higher than in the Alvar engine during a part of the exhaust stroke at 3000 rpm. Exhaust backpressure ≈ 1.05 bar. The two valves that were used in the Alvar engine is of the same size as in the standard 4 valve engine i.e. too small for a two valve engine. A better alternative would be to use the valves from the Volvo B5252 engine, this would probably eliminate any larger differences between the engines in terms of valve throttle losses.

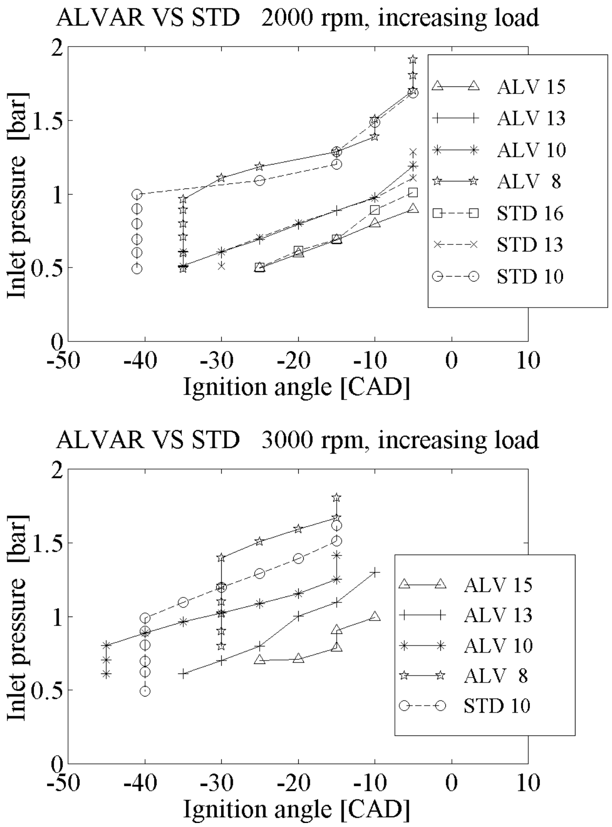


Figure 9. Inlet pressures and ignition angles.

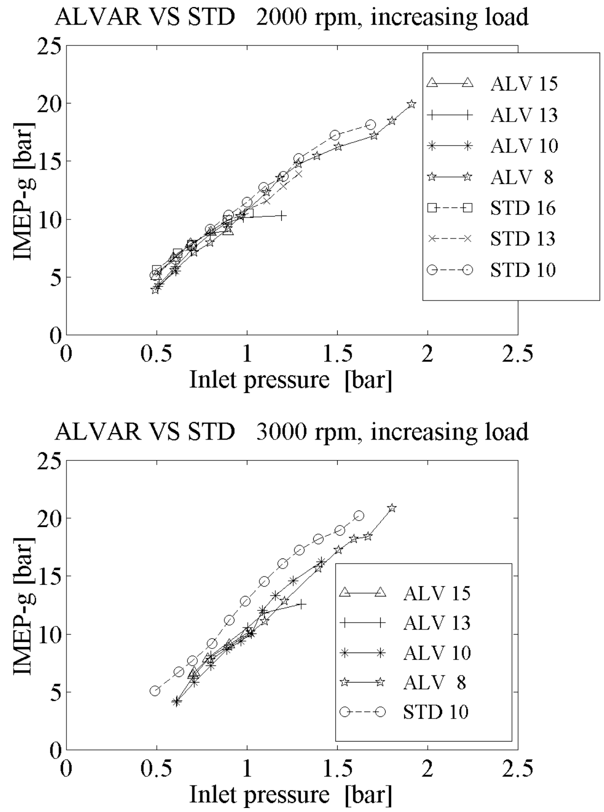


Figure 11. Gross indicated mean effective pressure.

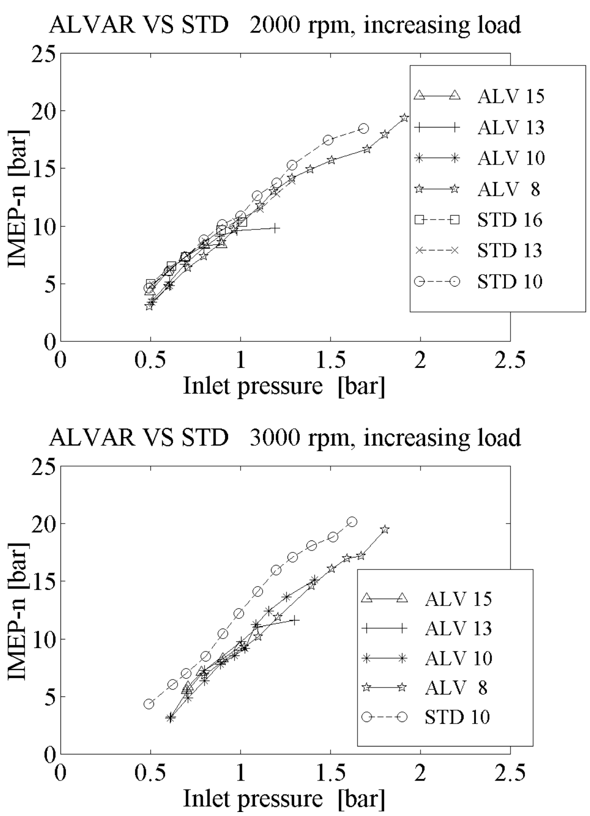


Figure 10. Net indicated mean effective pressure.

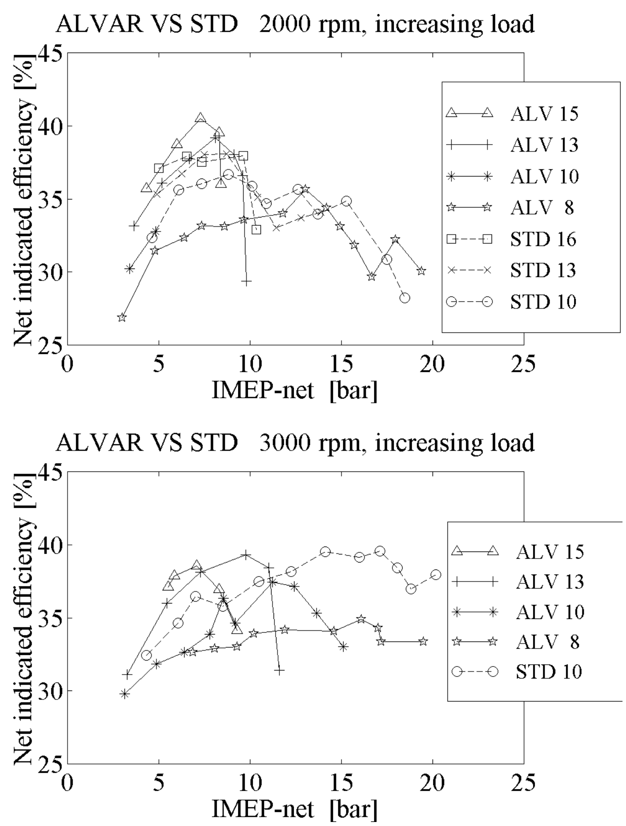


Figure 12. Net indicated efficiency.

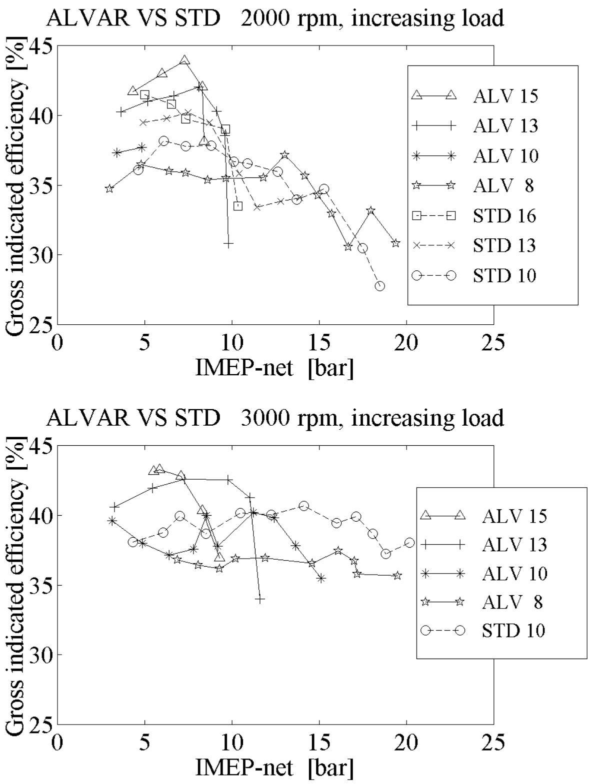


Figure 13. Gross indicated efficiency.

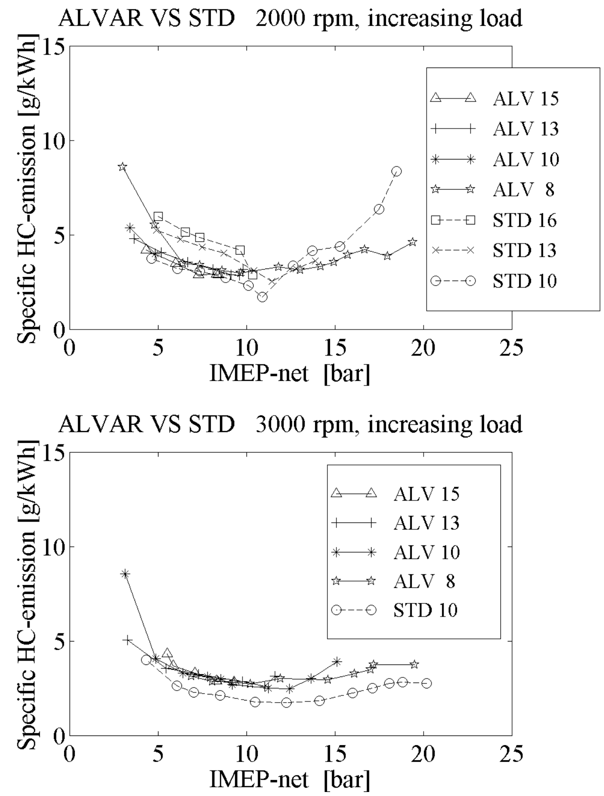


Figure 15. Specific HC - emissions. ($C_3H_8$ equivalent).

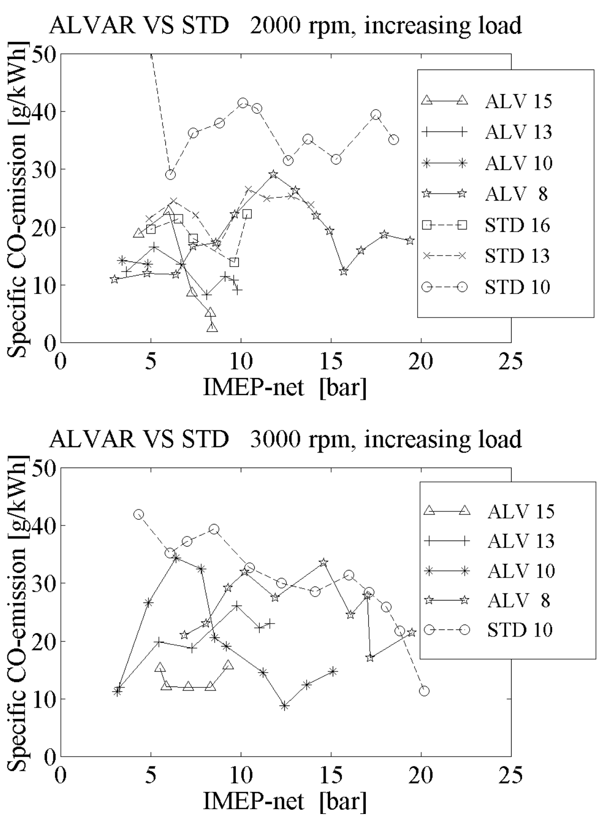


Figure 14. Specific CO - emissions.

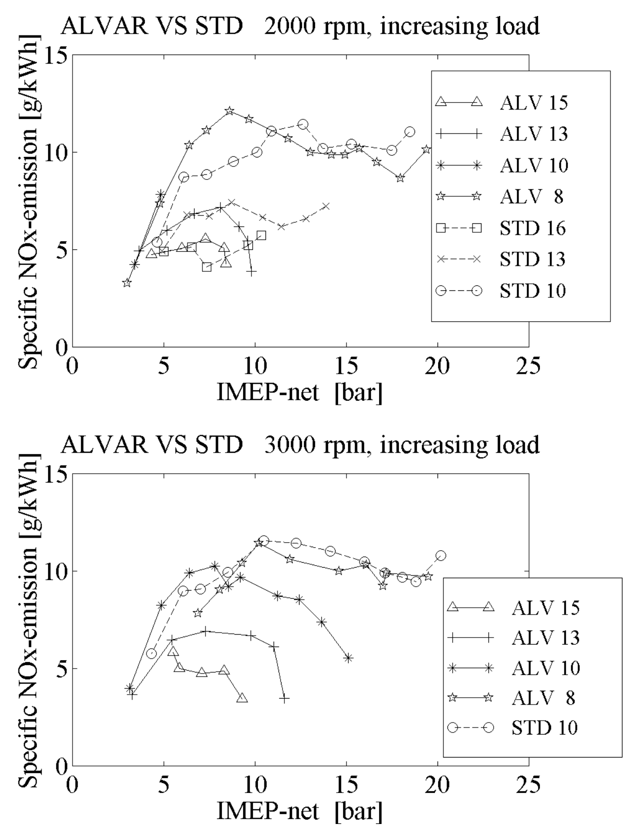


Figure 16. Specific NOx - emissions.

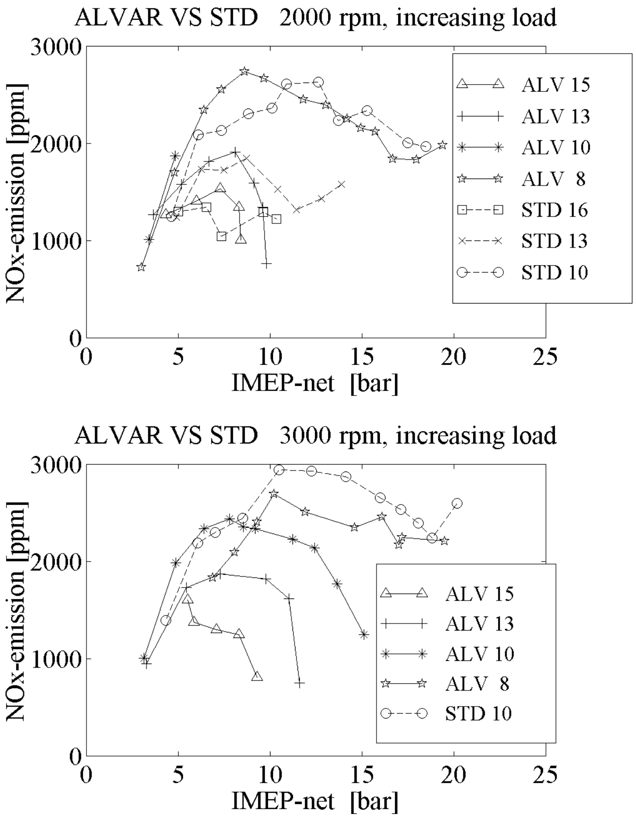


Figure 17. NOx - emissions, ppm.

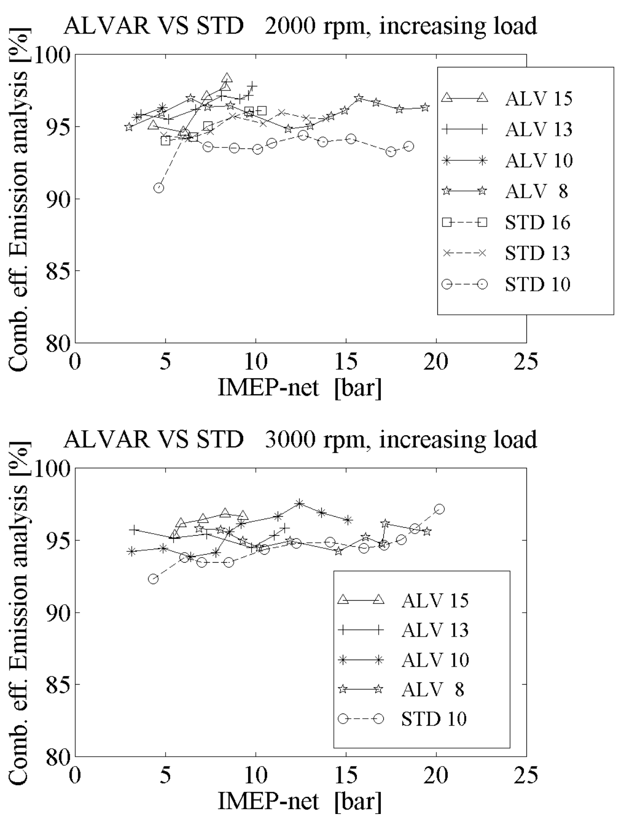


Figure 18. Combustion efficiency.

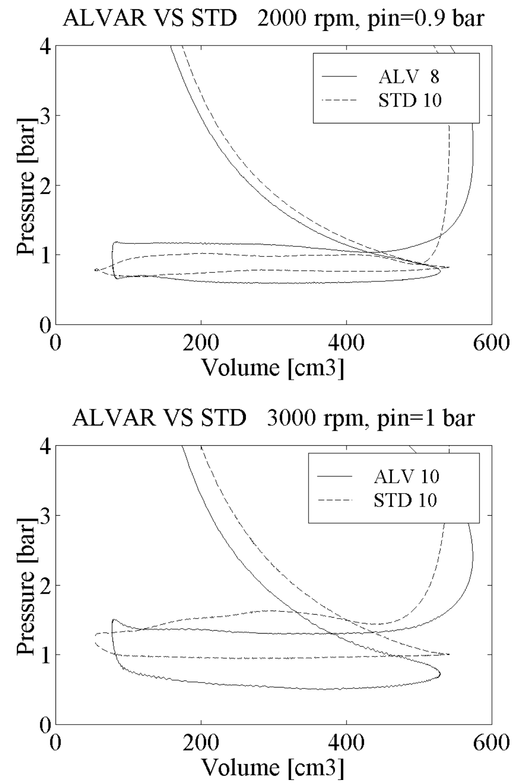


Figure 19. Intake and exhaust strokes.

## DISCUSSION

The Alvar engine can be a solution for variable compression ratio with potential applications in the automotive market. The main problem with the Alvar engine is that it uses two sets of cylinders and thereby needs two (different) sets of mechanisms. And then there is the phaseshift mechanism, although some variable valve timing (VVT) systems also use such mechanisms. VVT can be interesting for the Alvar engine. The Alvar engine can probably be combined with several VVT –systems but this will probably be more of a technical, rather than an economical interest.

The results indicate that at part load, high compression ratio, a relatively high increase in efficiency can be achieved and this without any direct increase in emissions. A further study in this field, and especially high compression ratio and its effect on EGR –tolerance together with EGR and its effect on the knock limit, would be very interesting. Information from such tests would be very valuable when optimizing an engine for efficiency.

The Alvar engine is far from being the only idea for variable compression ratio. There are lots of patents that have been filed on the subject but none of them has been commercially accepted yet [5]. Most of these solutions suffer from a complex and expensive design

## CONCLUSIONS

- Variable compression ratio renders a possibility for increasing the efficiency at part load conditions.
- At high compression ratio: It is not critical in terms of efficiency if the spark advance has to be reduced from MBT timing to avoid knock. It will bring the benefits of lower NOx – emissions.
- Variable compression ratio gives a possibility to increase the maximum power output through lower compression ratio and high boost pressure.
- The Alvar engine is a possibility in achieving variable compression ratio and it has shown to give the expected benefits of such a engine.
- The Alvar engine does not give higher HC-emissions than a standard engine, contrary to what was expected.

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4. J. B. Heywood; “Internal Combustion Fundamentals”, McGraw-Hill, New York 1989, ISBN 0-07-100499-8. Equations 3.27 and 4.69.
5. IBM Patent Server; http://www.patents.ibm.com, Search on “Variable compression ratio”.


## DEFINITIONS, ACRONYMS, ABBREVIATIONS

ALV Indicates results with the one cylinder Alvar engine.

STD Indicates results with the conventional four valve, one cylinder engine.

IMEP Indicated mean effective pressure

- Net Indicates that calculation includes exhaust and intake strokes i.e. pumping losses.

- Gross Indicates that calculation excludes exhaust and intake strokes.

MBT Maximum brake torque. Indicates the spark advance that gives the highest torque.

EGR Exhaust gas recirculation

TDC Top dead center. Piston position when the piston is at its uppermost (inner) position.

BDC Bottom dead center. Piston position when the piston is at its lowest (outer) position.

$\varphi$ Phaseshift angle.

$V_{max}$ Maximum cylinder volume at start of compression stroke.

$V_{min}$ Minimum cylinder volume at end of compression stroke.

$V_c$ Clearance volume (= Vmin).

$V_d$ Displaced or swept volume.

pin Manifold air pressure (Inlet pressure)

$r_{c,nominal}$ Nominal compression ratio (NCR).

$r_{c,effective}$ Effective compression ratio (ECR).