

STIRLING-ENGINE IN A NOVEL ALPHAGAMMA CONFIGURATION – A KEY FOR MAINTENANCE-FREE OPERATION

Josef Frauscher^{1,*}, Franz Diermaier¹, Hans-Jürgen Brandt¹ and Michael Gschwendtner²

¹Frauscher Thermal Motors GmbH, Gewerbestraße 7, 4774 St. Marienkirchen, Austria

²Auckland University of Technology, 55 Wellesley St E, Auckland, New Zealand

Abstract. Since 2001, Frauscher Thermal Motors have been conducting research in the field of thermodynamic machines, in particular Stirling engines of various types. One important development step is the invention of a Stirling engine in an alphagamma® configuration. In this configuration, the expansion piston is designed as a differential piston with its ring surface connected to the cold volume.

In this paper, the design advantages of the alphagamma® configuration in comparison with a traditional alpha configuration are shown analytically by using a polytropic model as a modification of the ideal adiabatic analysis. The findings were confirmed by also simulating the proposed alphagamma® configuration in a Sage model which was validated against experimental data with very good agreement.

The results of both methods show that the counter-productive compression work can be reduced to almost zero – which makes the compression piston a displacer and explains the name alphagamma® – with the expansion work also reduced for the same net work output. As a consequence, the forces on the pistons, and thus, on the bearings can be significantly reduced, also leading to smaller piston side-loads, less friction and wear. The combination of all advantages allows the design of a mechanically sound and inexpensive machine.

1 Introduction

When we created the first strategies and calculations for a marketable Stirling engine in 2001, we initially went along with the state of the art of that time. **Fig. 1** shows several alpha-, beta- and gamma-engine prototypes in a power range of 0.5 to 11 kW, which we developed during our research work. Nevertheless, to us, none of these prototypes seemed promising for a possible market entry.

This paper describes a major improvement for Stirling engines, relying on our own experience that we have been able to make for more than 20 years of research. In order not to go beyond the scope of this article, we limit the data to engines based on the alpha principle (single-acting) and analyse the effects of variations in the compression volume and the phase

* Corresponding author : josef.frauscher@frauscher-motors.com

angle on the piston forces and negative work per revolution. We use engine dimensions that we have tested for years with similar values on built prototypes. This allows numerous comparisons with measured parameters and provides security in terms of the quality of the information given.



Fig. 1. Prototypes of Frauscher Thermal Motors Stirling engines

Comments on the mechanical design of an alpha-Stirling

Developers of Stirling engines are undoubtedly well advised if they keep the design generally simple, robust and all parts as multifunctional as possible. For example, it is advantageous to use the crankcase as a buffer space and as a housing for the generator. Such a design prohibits the use of lubricating oil in any case, since it cannot be kept away from the process space permanently via the piston rings. The use of lifetime lubricated rolling bearings is state of the art and with sufficient leeway to their limit load, lifespans of tens of thousands of hours can be achieved. If the generator is arranged in the area of the crankcase, there is no need for a pressure-tight and friction-intensive rotary feedthrough for the crankshaft to the outside and thus a sensitive wear part of the machine.

A significant simplification is achieved if the phase angle between the pistons can be chosen to be 90° . This allows a most easy mass balance by simply placing counterweights directly at the opposite of the crank pin. The prerequisite for this is that the mathematical product of the piston mass and the crank radius is identical for both pistons.

It is also advantageous to choose a small connecting rod ratio (λ), i.e. a relatively long connecting rod in relation to the crank radius. This not only reduces the piston side-loads, but also spares the mass balancing of the second order mass forces.

One of the greatest challenges are sufficiently long-lasting dry-running piston guides. Here it is important to find a balanced interaction between surface hardness and roughness of the cylinder wall with a suitable plastic compound on the piston skirt. Although some experience can be used from the field of oil-free piston compressors, the thermal and mechanical load conditions in a Stirling engine are widely different from those in compressors. In addition, the process gas helium or hydrogen, for example, behaves completely differently in the "lubricating gap" than conventional air in compressors.

Of course, when considering dry-running guides, the principle applies that the specific surface load between the piston skirt and cylinder wall must be kept as low as possible for

the target of low wear and a long service life. Since the total area is very limited in terms of a compact design, measures to reduce the piston side-loads are essential for a satisfactory maintenance-free operating time of the Stirling engine. It should also be noted here that low piston side-loads not only reduces wear, but also the friction losses in the sense of a high overall efficiency.

Measures to reduce piston side-loads are:

- A low connecting rod ratio (as mentioned above)
- Reduction of the pressure amplitude for minimal piston forces

The following chapters describe ways to reduce the piston forces without having a negative effect on the engine's performance.

2 Calculations

2.1 The complex gas-process

It is known that the changes of state of the working gas in a Stirling engine are highly complex. While an approximately isothermal behaviour can be expected in the regenerator, the processes in the cooler and heater heat exchangers are quite polytropic and more adiabatic in the cylinder chambers. Several calculation programs simplify the consideration to isothermal processes in all sections, which can lead to considerable errors in the results. Frauscher Thermal Motors therefore uses a polytropic simulation model including heat transfer and pressure drop, which is a further development of the ideal adiabatic analysis [1]. It comes very close to the actual conditions in a real Stirling cycle, which significantly increases the calculation accuracy. The following pressure-volume diagram shows a comparison of the results of an isothermal, adiabatic and polytropic simulation with actually measured values of a test bench prototype. Measurement position of the process pressure is between compression space and cooler entrance. As **Fig. 2** shows, the polytropic simulation provides a practical approximation of the measured data and therefore is used for the upcoming simulations and comparisons.

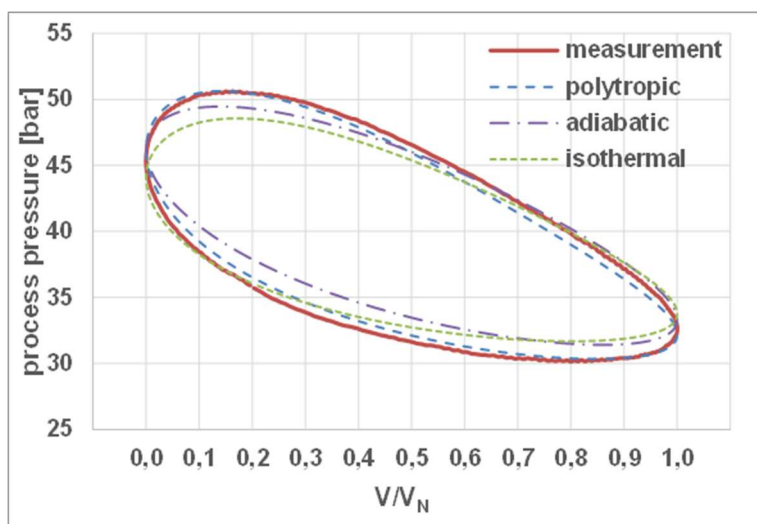


Fig. 2. Measured and calculated pressure-volume diagrams of a prototype engine

2.2 Optimization

An alpha-type engine with a constant expansion stroke volume, constant heat exchanger volumes but variable compression volume and phase angle will be considered, with the net work per revolution being kept the same in every variation by adjusting the mean process pressure. The calculations are extended to a variant with differential piston (alphagamma® process) in order to obtain a direct comparison with alpha-type machines. A cooler heat exchanger temperature of 300 K and a heater temperature of 900 K are assumed for all variants. The results should reveal the value of the cycle-work of the individual pistons per revolution, the total positive and negative work of both pistons and the piston forces. **Fig. 3** shows schematically the examined alpha-type machine with its fixed and variable sizes.

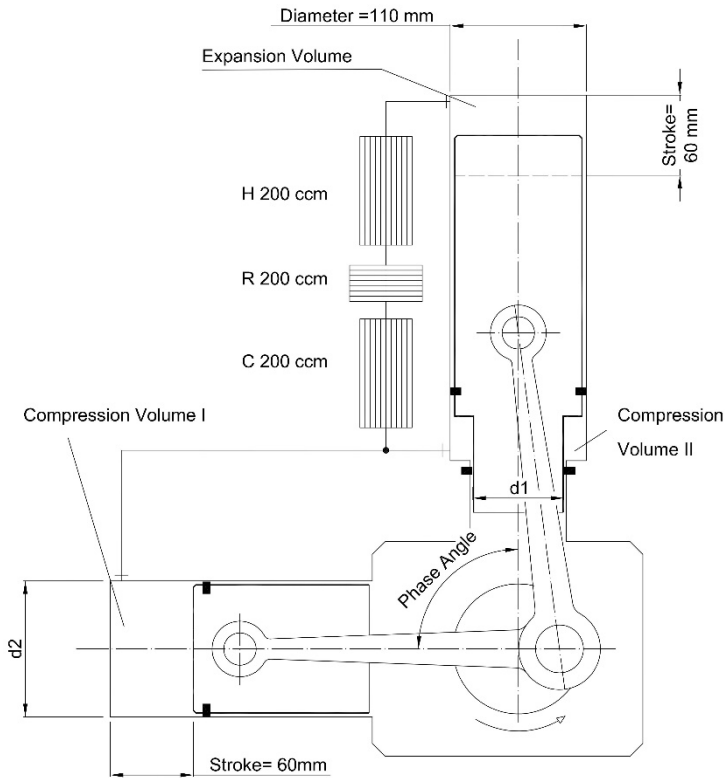


Fig. 3. Schematic of an alpha/alphagamma® type engine with partly fixed and variable dimensions

2.3 Variants

Table 1 shows the investigated variants, their stroke volume and phase angle. In addition, the proportion of the total compression volume in relation to the expansion volume is listed.

Table 1. Summary of volume and phase angle variations

	displaced volume	% of expan- sion volume
Variant a: standard alpha		
Compression volume I (d ₂ =110 mm)	570 cm ³	100%
Compression volume II (d ₁ =110 mm)	0 cm ³	
Expansion volume	570 cm ³	
Phase angle	90°	
Variant b: optimized alpha		
Compression volume I (d ₂ =90 mm)	382 cm ³	67%
Compression volume II (d ₁ =110 mm)	0 cm ³	
Expansion volume	570 cm ³	
Phase angle	135°	
Variant c: alphagamma®		
Compression volume I (d ₂ =80 mm)	302 cm ³	
Compression volume II (d ₁ =75 mm)	305 cm ³	
Resulting compression volume	429 cm ³	75%
Expansion volume	570 cm ³	
Resulting phase angle	135.3°	

2.3.1 Variant a: standard alpha process

This is a classic alpha configuration with a V-shaped cylinder arrangement, 90° cylinder angle (i.e. phase angle) and the same stroke volume in both cylinders. Since the bottom diameter of the stepped piston is theoretically the same as the large diameter (d₁=110 mm), there is no annular volume. Thus, this variant corresponds to a typical alpha configuration as it can often be found in publications and in specialist literature. In several cases similar machines have been built and put on the market.

If the indicated work over one crankshaft revolution, as shown in **Fig. 4**, is considered, then high proportions of negative work can be seen, which have to be compensated by even higher positive sections in order to obtain the desired net work. As shown in the calculations below, such a work process generates extremely high pressure strokes along with the highest piston forces. Further disadvantages such as high torque fluctuations, which lead to high current oscillations in generators, as well as comparatively high adiabatic losses would be suitable subjects for further examinations.

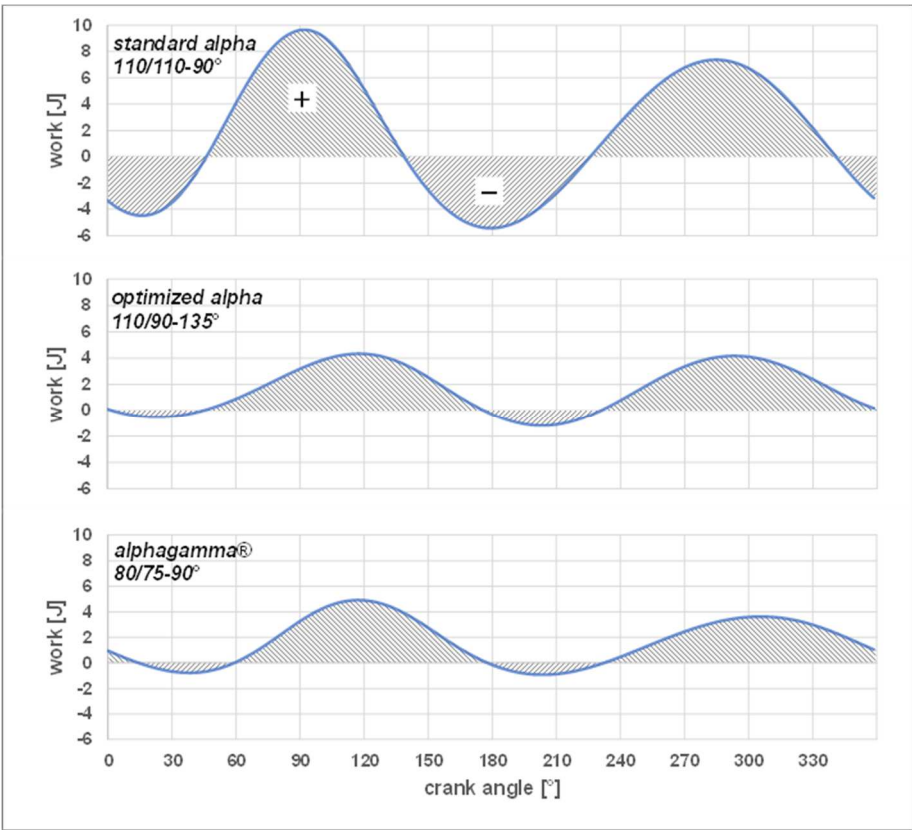


Fig. 4. Comparison of the indicated process work per cycle of the three variants

2.3.2 Variant b: optimized alpha process

The research activities of Frauscher Thermal Motors early revealed the insight, that a design according to variant a) leads to unsatisfying results. A series of tests over several years showed that optimal phase angles between 120° and 140° and a compression volume that is only 65-75% of the expansion volume result in a significant reduction in the amount of negative work per revolution. Ideal values, of course, have to be individually adapted and depend on the temperature ratio, the volume of the heat exchangers and their efficiency. An engine design with the type designation “A600” was manufactured in several versions and successfully operated in internal and external applications. The work diagram, shown in **Fig. 4**, clearly shows that the negative work contribution decreases to less than 10%, based on the positive work.

The required mass balance for this type of alpha-engine proved to be complex, as it could not be carried out entirely by simple crankshaft counterweights. The search for simpler designs, in addition to the goal for higher levels of efficiency finally led to the alphagamma® solution.

2.3.3 Variant c: alphagamma® process

In this variant, the expansion piston consists of a stepped piston with the typical expansion volume above and a ring-volume below it. The compression piston has, as in variant a), a

phase offset of 90° (in the sense of a simple mass balance of 90° two-cylinder piston engines). The effective compression volume is hereby divided into two compression spaces. With the ring-volume (compression volume II) having a phase offset to the expansion volume of 180°, together with compression volume I, the total compression volume leads to a resulting phase angle of 135.3°. **Fig. 5** shows the trigonometric relationship of the alphagamma® process in a vector diagram. The expansion volume, the two compression volumes as well as the size and phase angle of the resulting compression volume can be seen.

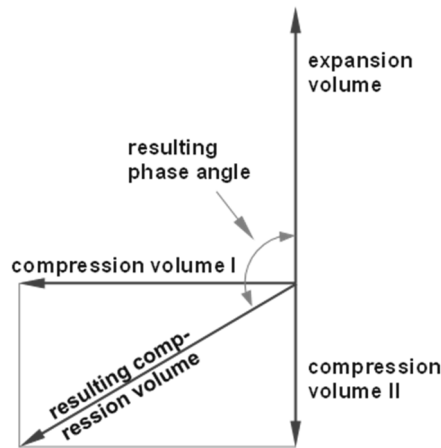


Fig. 5. Vector diagram of phase angle and resulting compression volume in the alphagamma® process

By looking at the process work in **Fig. 4** and comparing the optimized alpha process with the alphagamma® process, no further advantages beyond simple mass balancing are initially recognizable. This is confirmed by the work values listed in the following **Table 2**.

Table 2. Sum of the positive and negative areas of the cycle-work

	standard alpha 110/110-90°	optimized alpha 110/90-135°	alphagamma® 75/80-90°
total work	590.6 J	590.5 J	591.1 J
positive work	1085.2 J	648.4 J	646.4 J
negative work	- 494.6 J	- 57.9 J	- 55.3 J
Ratio (pos./neg.)	45.6%	8.9%	8.6%

2.4 The effect of the stepped piston

The additional advantages of the alphagamma® process only become apparent when the work of the individual pistons and their forces are calculated separately. **Fig. 6** shows that in the standard alpha process, the expansion piston has to compensate the negative work of the compression piston. Only the additional work of the expansion piston can be used as net work. With the alphagamma® process, on the other hand, the compression piston even contributes a small positive amount to the net work done, while the work by the expansion piston is extremely reduced.

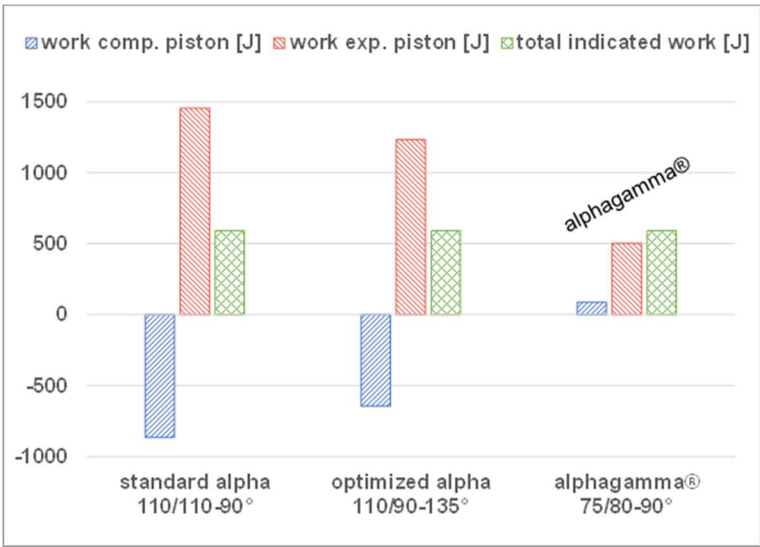


Fig. 6. Piston work per cycle

The individual piston work is directly connected to the calculated piston forces. **Fig. 7** shows the maximum gas force on the pistons. It is taken into account that the gas pressure in the buffer space acts as a constant mean pressure on the bottom of the pistons and in this respect only the differential pressure was used to determine the piston force. In addition, the simulation method used, also takes into account the pressure losses across the heat exchangers and the regenerator, based on an engine speed of 1000 rpm.

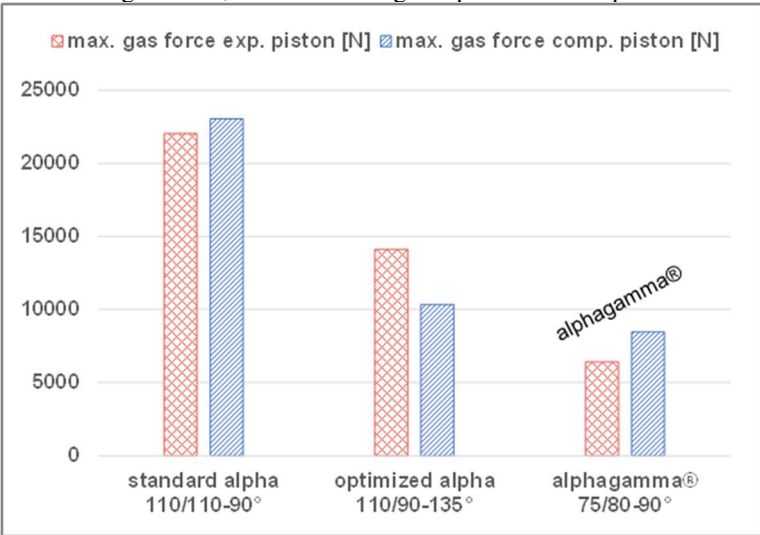


Fig. 7. Maximum gas pressure forces on pistons

The actual piston force that acts on the connecting rod can only be represented by including the mass force. Here, another advantage of the alphagamma® process comes to light: assuming a piston mass of 4.5 kg for each of the two pistons and a speed of 1000 rpm, which has been tried and tested in our example, the mass force hardly increases the maximum piston forces. In the case of the expansion piston, the gas force dominates anyway, which has its maximum at approximately the middle of the stroke. Looking at the compression piston,

the process pressure has its maximum at around the top dead centre and its minimum pretty much at its bottom dead centre. Since this counteracts the mass force, the maximum piston force is again significantly reduced. **Fig. 8** shows the difference in comparison to **Fig. 7**.

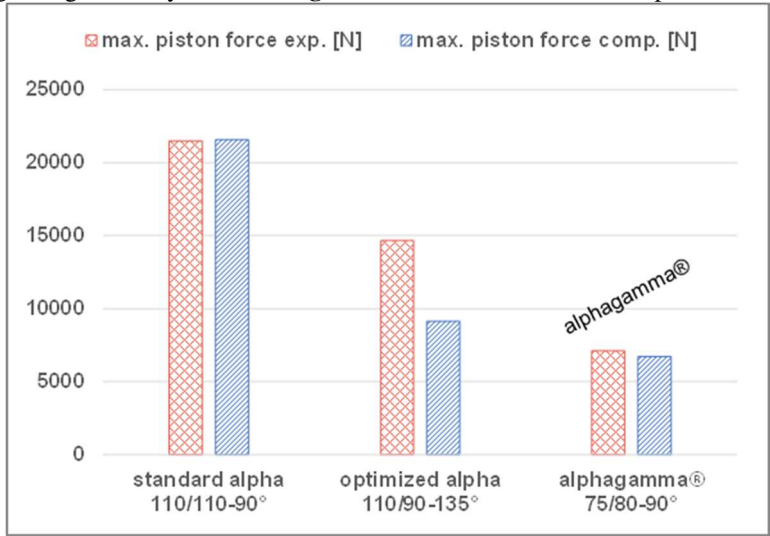


Fig. 8. Maximum total piston forces (mass-force included)

Table 3 shows the results in absolute values. It is shown that the use of a stepped piston as an expansion piston with suitable dimensions not only significantly reduces the negative work per cycle and the associated compression loads on the machine, but also the individual work of the pistons and their maximum forces. At an expansion and compression volume of 570 cm³ each, a phase angle of 90° (standard alpha process), and when a practical piston mass is included, the maximum piston force of the expansion piston is 3 times higher than in the alphagamma® process. The force on the compression piston is even about 3.2 times as high.

Table 3. Simulation results

	standard alpha 110/110-90°	optimized alpha 110/90-135°	alphagamma® 75/80-90°
mean process-pressure	33 bar	54.5 bar	50.5 bar
max. piston force comp.	21573 N	9142 N	6718 N
max. piston force exp.	21482 N	14650 N	7100 N
work compression piston	-863.6 J	-642.7 J	87.7 J
work expansion piston	1454.2 J	1233.2 J	503.4 J
total work	590.6 J	590.5 J	591.1 J

The alphagamma® process paves the way for lubrication-free dry-running technology, due to the low piston side-loads. Additionally, a simple, inexpensive and reliable engine design for long maintenance-free intervals can be achieved. The data for the "standard alpha", "optimized alpha" and "alphagamma®" variants are not only based on theoretical calculations, but have also been verified by test bench results.

3. Sage Simulations

3.1 Simulation software Sage

An alphagamma® Stirling engine was modelled in the commercial Stirling simulation software Sage written by David Gedeon [2]. The physical dimensions of all components are similar to variant c) (see 2.3.3.) and were based on an actual alphagamma® Stirling engine prototype. This included the piston strokes as well as the mean pressure the engine is operated at. In its ability to model heat transfer processes and fluid friction of oscillating flow – even through porous media such as the regenerator matrix – Sage can be regarded as a very powerful 3rd order model. Each component of the system is analysed in a one-dimensional form, subdivided into a user-defined number of spatial increments. While Sage primarily models the thermodynamic behaviour of the working gas, it also takes into account the interaction of the gas with the adjacent walls, as well as heat transfer by conduction in the canister walls in both axial and radial directions. In this respect, Sage is somewhere between a one- and a two-dimensional modelling tool. Sage operates in a frequency domain and solves all fundamental conservation equations for each cell until it converges to a solution. **Fig. 9** depicts what the modelled alphagamma® configuration looks like in Sage.

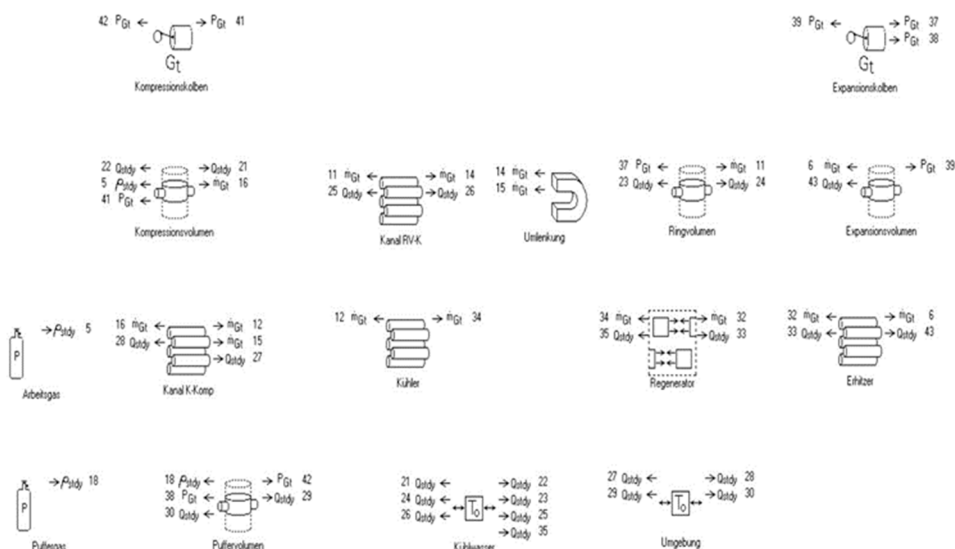


Fig. 9. Screenshot of the Sage simulation model of the Frauscher alphagamma® Stirling engine

3.2 Validation of the Sage simulation

In order to validate the Sage model, a comparison was made between the experimental p-V data and the Sage simulation (**Fig. 10**). A few assumptions had to be made for the simulation:

- Since the minimum and maximum volumes are kinematic constraints independent of any simulation, and since there is inevitable inaccuracy in the determination of the actual gas volume in a real engine, the volume was adjusted in Sage by adding a small amount of dead volume in the regenerator canister.
- Another minor adjustment was required for the average gas pressure. While Sage keeps the average gas pressure as specified by the user constant by adjusting the gas mass, in a real engine, the average gas pressure slightly increases after start-up.
- The volume for the buffer space was chosen such that that the pressure amplitude matched the measured one.
- Other than that, only the heat source and heat sink temperatures were chosen to be the same as in the experiment.

As can be seen in **Fig. 10**, the measured p-V data and the simulation results obtained with Sage agree very well. Sage underestimates the p-V area by only 6%. It is unclear, however, how large the experimental inaccuracy is. It should be noted that the agreement between simulation and real prototype is almost ‘spot-on’ during the expansion phase (top of the p-V loop), while there is more discrepancy during the compression phase (bottom of the p-V loop). An explanation as to why this is cannot be given at this stage. Furthermore, Sage predicts a net power output of 8864 W at a speed of 1007 rpm that the prototype was tested at. The measured electrical power output was 6674 W, which would indicate that approx. 25% of the indicated power is lost due to mechanical friction and in the mechanical to electrical conversion process.

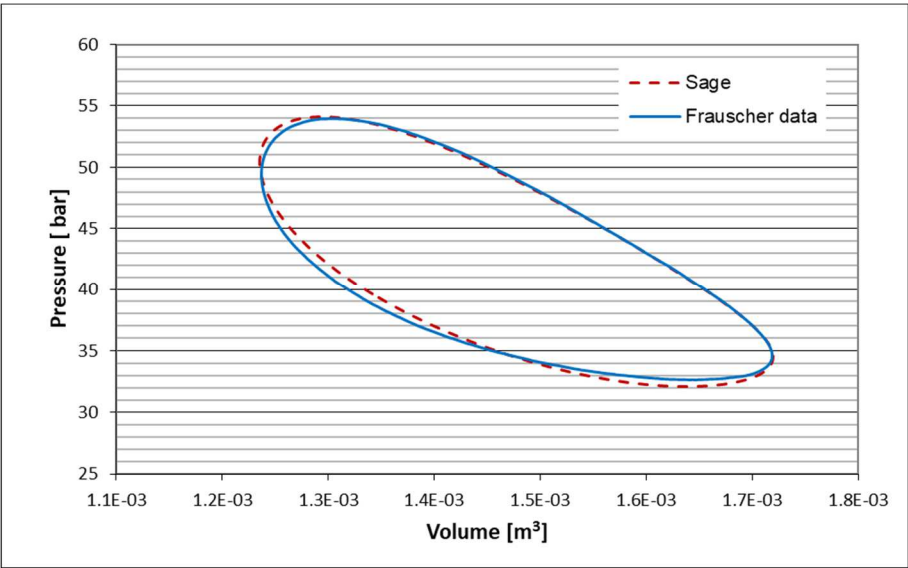


Fig. 10. Comparison between the experimental p-V loop and Sage results

Another encouraging validation of the Sage simulation is the comparison with some temperature measurements. **Table 4** shows the measured temperatures in three locations (average over 30 minutes; 180 values) and the corresponding Sage results (average over one cycle). With the exception of the hot gas temperature at the inlet to the expansion cylinder

where the difference is approx. 15 K, the remaining gas temperatures are almost identical. While this extremely close agreement in those two cases is most likely coincidental, it is still indicative of how well Sage captures the prevailing heat transfer mechanisms.

Table 4. Comparison of gas temperatures between the experiment and Sage

Location	Experiment	Sage
Hot gas temperature at inlet to expansion cylinder	700.1 °C	684.8 °C
Hot gas temperature at inlet to regenerator	724.4 °C	724.3 °C
Cold gas temperature at inlet to compression cylinder	50.5 °C	50.7 °C

4. Results and Discussion

4.1 Volumetric phase relationship

Probably the most obvious effect the proposed configuration has on the thermodynamic behaviour compared to a pure alpha configuration is on the volumetric phase relationship between expansion and compression space. Since the compression space in the alphagamma® engine is now connected to two piston faces that are moving out of phase by 90°, the combined effect changes the phase difference between the total volume and the pressure, which is critical for the p-V work in terms of the integration of pdV over a complete cycle. Fig. 11 shows the cyclic change of the total volume and the pressure over a complete cycle for both the alpha configuration and the alphagamma® engine. In both cases, the pressure slightly lags behind the minimum volume. By inspection of the numerical values in Excel, it was found that the pressure lags by only 20° in the alpha configuration, while it lags by about 40° in the alphagamma® engine. What effect this has on the work done on and by the gas on the pistons can be seen in the following p-V power diagrams.

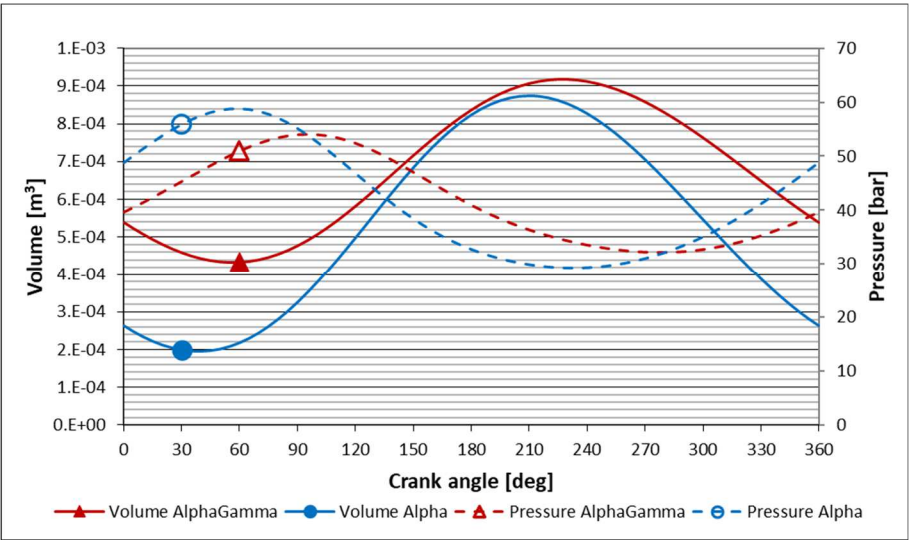


Fig. 11. Comparison of total volume and pressure between alphagamma® and Alpha

4.2 p-V power

Fig. 12 shows the cyclic p-V power of the working gas in the expansion space (red), the annular space created by the stepped piston (green), the compression space (blue), as well as the total p-V power of the entire gas space in a comparison between the alpha configuration (dashed lines) and the alphagamma® (solid lines). It should be noted that the Sage model of the alpha configuration was created by simply removing the annular gas space (by removing the step in the piston) and the duct connecting the annular space with the compressions space in the alphagamma® Sage model. As a result, the alpha engine has approx. 8% less dead volume and an almost 40% higher pressure amplitude. However, the alpha engine produces a net power of only 8592 W, while the alphagamma® generates 8868 W. According to the Engineering Thermodynamics sign convention, a positive value in **Fig. 12** means that the gas performs work on its boundary. Since the expansion piston is at top dead centre for a crank angle of 0°, the p-V power has just passed zero and increases in positive direction as the piston moves down and the expansion space becomes larger (red lines). The opposite is true for the annular gas space in the alphagamma® (green line) as the two piston faces are out of phase by 180°.

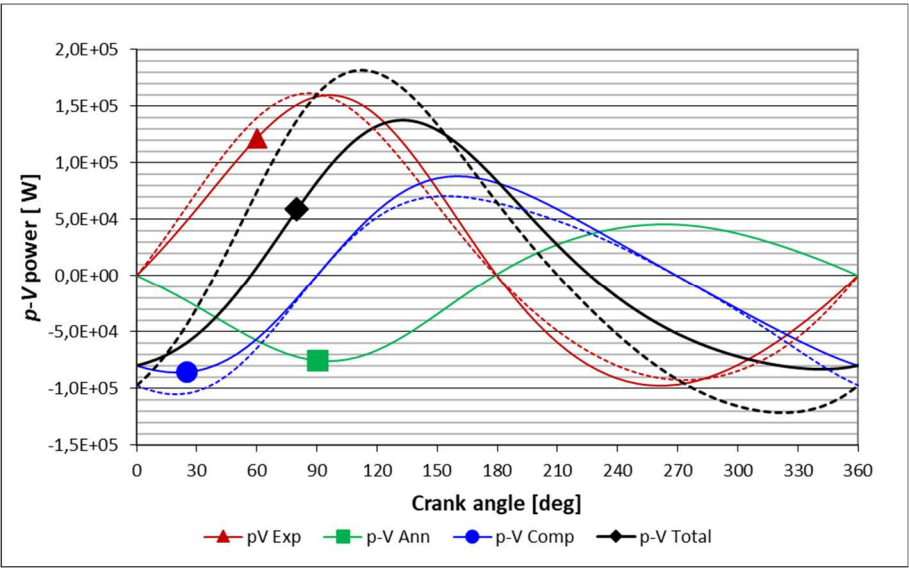


Fig. 12. Cyclic p-V power (at 1007 rpm) done by the gas in various space for both the alpha configuration (dashed lines) and the alphagamma® (solid lines).

While the cyclic p-V power of the expansion space is very similar for both the alpha engine and the alphagamma®, the p-V power of the compression space is clearly shifted in positive (i.e. power-producing) direction in the alphagamma®. This manifests itself in the compression piston in the alphagamma® requiring no work input, in fact, even producing a small amount of power output as can be seen in **Table 5** (highlighted in bold). In this respect, the compression piston almost behaves like a displacer as in a gamma configuration. The values in **Table 5** clearly show that the expansion piston in the alphagamma® engine does both the expansion and compression work on the gas, resulting in less than half the net power as in the alpha configuration. Consequently, this results in much lower piston, and thus, bearing forces as will be discussed below.

Table 5. Net power of compression and expansion piston in both the alpha and the alphagamma® engine

	Alpha	alphagamma®
Net power compression piston	-9649 W	138 W
Net power expansion piston	18240 W	8730 W
Net power produced by the engine	8591 W	8868 W

Fig. 13 shows the cyclic p-V power as a sum of the various gas spaces including the buffer space for both the alpha configuration (dashed line) as well as the alphagamma® engine (solid line). It can be clearly seen that the negative power is significantly reduced in the alphagamma® configuration.

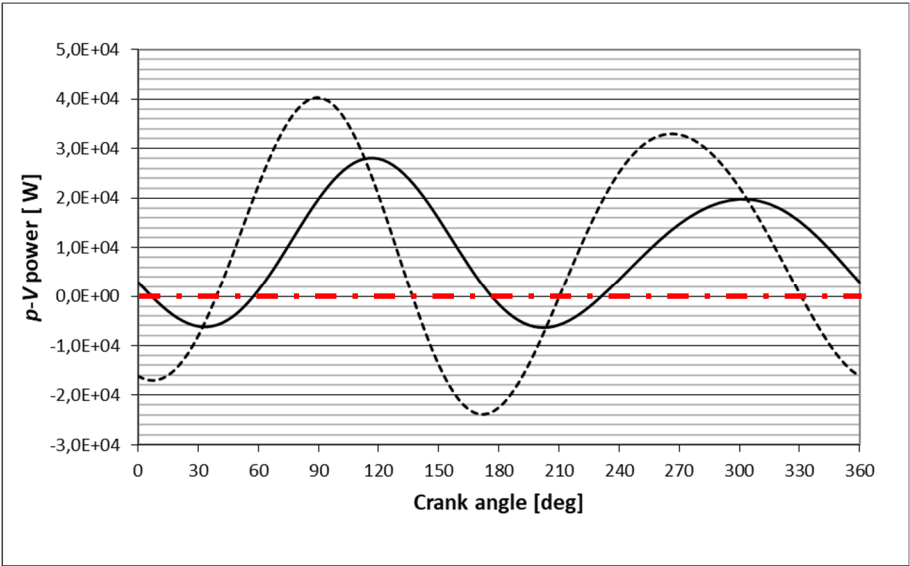


Fig. 13. Total cyclic p-V power (at 1007 rpm) done by the gas for both the alpha configuration (dashed line) and the alphagamma® (solid line).

4.3 Piston forces

Fig. 14 shows the cyclic ‘required forcing function’ for the compression and expansion piston for both the alpha configuration as well as the alphagamma® engine. In Sage, the ‘required forcing function’ is the force that is required to execute the prescribed motion, taking into account inertia forces due to acceleration as well as the instantaneous gas forces. Based on the previous discussion of the cyclic p-V power in various gas spaces of both engines, it is not surprising that the force amplitudes are much smaller in the alphagamma® engine. By inspection of **Fig. 14** it can be seen that the maximum occurring piston force in the compression piston of the alpha configuration is approx. 1.7 times higher than in the alphagamma® engine. The difference in the expansion piston is even higher with an almost three times higher force in the alpha configuration compared to the alphagamma® engine.

This means that there is a significantly smaller load on the bearings in the proposed alphagamma® configuration. The reduced forces on the piston also mean that the side-loads on the piston seals are much lower, which results in less friction and less wear of the seals.

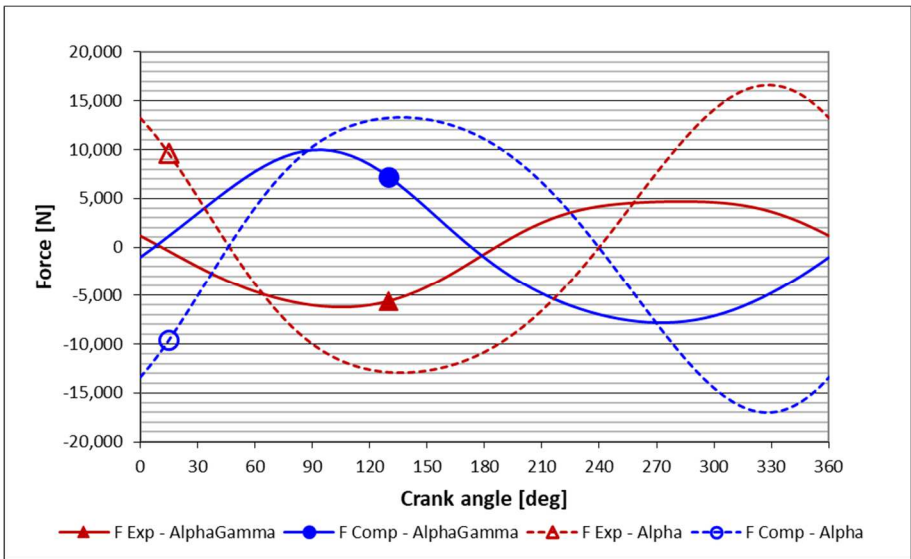


Fig. 14. ‘Required forcing function’ for the compression and expansion piston in both the alpha configuration and the alphagamma® engine

5. Conclusion

The simulation results of the novel alphagamma® Stirling engine configuration show, that the use of a stepped expansion piston with suitable dimensions not only significantly reduces the negative work per cycle and the associated compression loads on the machine, but also the individual work of the pistons and their maximum forces.

A simulation of the alphagamma® engine in Sage also reveals that the piston forces can be substantially reduced by using a stepped expansion piston in an alpha configuration and connecting the annular gas space to the compression space. For the same net power output, the maximum occurring piston force of the compression piston can be reduced by over 40% compared to a standard alpha configuration.

By executing the Stirling cycle between three piston faces that are all out of phase, a phase shift between the overall gas volume and the pressure can be achieved. In this particular case, this leads to a positive net p-V power done by the compression piston of up to 10% of the total work. The opposing gas forces on the expansion piston result in a much smaller net force.

We performed several tests with a new Stirling engine with the alphagamma® configuration on our test bench with a gas-fired burner. We could achieve an electrical efficiency over 32% (based on net calorific value) and an electrical output between 6 kW and 7.5 kW depending on the load pressure of the working gas between 40 and 43 bar Helium and the temperature on the hot side between 700°C and 740°C. Frauscher Thermal Motors has been performing life tests with eight engines on internal and external testing sites having accumulated 35.000 hours of operation by now.

The alphagamma® process paves the way for lubrication-free dry-running technology, due to the low piston side-loads. Additionally, a simple, inexpensive and reliable design of the machine for long maintenance-free intervals can be achieved.

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