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Stirling Engines with alphagamma® Procedure

Author: Josef Frauscher
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Characteristics and perspectives of the new engine technology

The need for decentralised and controllable electrical energy generators is undisputed. Biogenic fuels are in the foreground in this regard, as their use can help reduce greenhouse gas emissions. This technology is expected to be revitalised through a newly developed process in Stirling engines, especially because it allows for the use of fuels outside of the strictly regulated fossil fuel grades.

Image: Frauscher

Introduction

Overshadowed by the internal combustion engine, Robert Stirling's invention has to this day been deemed insignificant. This Scottish pastor's ingenious design, dating back to 1816, actually deserved more. Why have Stirling engines had so little commercial success? The author has been involved with this technology for over 30 years and spent considerable resources on the research and development of Stirling generators within the framework of his company for 20 years. On this basis, the author attempts to get to the bottom of this question.

To a certain extent, the impetus to write this article came about because successes had outweighed setbacks in the research work. A more significant factor, however, was the worrying development of the fossil fuel market coupled with the volatility of booming solar and wind renewable energy sources. In other words: Decentralised, controllable aggregates for the use of biogenic fuels as well as applications for balancing volatile energy sources are needed now more than ever. The Stirling engine offers a solution for both challenges and has the capacity to make a valuable contribution to the future energy landscape. This aggregate is suitable for use in grid interconnections as well as applications of reliable island supplies or to cover grid outages.

In this paper, we attempt to determine one of several possible reasons for the failure of Stirling engine developments; which are often very cost-intensive. In doing so, we draw on our own experience gained in the course of 20 years of research. To stay within the scope of this paper, we limit our detailed analysis to engines based on the alpha principle (single-acting) and analyse the effects of variations of the compression volume in connection with the phase angle on the piston forces and the negative work per revolution. This is carried out using dimensioning that has been tested for years using similar values on completed machines. This allows numerous comparisons with measured parameters and provides certainty in the result quality.

On the shadowy existence of Stirling engines

To get to the bottom of why the market penetration of the Stirling engine was sluggish, there is no need to recount in detail the technical developments since its invention; plenty of literature and internet content has already been published in this regard. Instead, it is sufficient to summarise the progress made era by era – in parallel with changes in society. For example, servicing engines for electricity generation on a daily or even hourly basis would be unthinkable today. In the past, "machine operator" was the specific job title of the person who, with oil can and spanner, worked to ensure sufficient availability of the machines at all times. Nowadays, when the aggregates are used as combined heat and power units, they are expected to go several thousand hours between services. These scales have now become a matter of course for heat pumps and refrigeration machines.

Unfortunately, Robert Stirling's intention that the Stirling engine would take the place of the accident-prone steam engine never came to pass at the time. However, a lively market for small machines in the 0.5 to 1.5 kW range, used in water pumps and other drives, developed in England and America in the second half of the 19th century. More powerful designs, which were operated using atmospheric medium pressure, failed due to the huge displacement required for the engines. The Ericsson Caloric Engine with 4 cylinders, each 4.26 m in diameter, may be a representative example.

What could be achieved from machines using high process medium pressure was only demonstrated by Philips in the mid-20th century. The considerable research work carried out up to the 1980s – including collaboration with NASA and the American automotive industry – resulted in astonishingly powerful machines whose performance closely approximated that of internal combustion engines.

Extreme levels of exhaust pollution from petrol and diesel engines had led to suffocating conditions in cities. This in turn provided an impetus for trials in automotive applications. However, despite the clean exhaust gases from its atmospheric combustion, the Stirling engine could not displace the internal combustion engine, especially since the introduction of the exhaust catalytic converter brought about a significant improvement in the air situation. The reasons why these efforts were discontinued included higher manufacturing costs and the Stirling engines' sub-optimal performance dynamics.

When topics like the environment and renewable energies came increasingly to the fore around the turn of the millennium, a fresh attempt was made. This was the time when the frequently occurring Stirling conferences featured lectures by many companies proposing extremely interesting solutions.

Unfortunately, only a few companies managed to secure commercial success. Considering the significant research funds invested by the entire scene, there is no doubt that the resulting harvest could only be deemed meagre. The successful models included two free-piston machines and one with an oil-lubricated kinematic crank mechanism. These definitely have one feature in common: They are based on a development time of several decades, with the baton periodically being passed to new owners or groups of companies willing and able to provide the required financial commitment.

In light of the recent development history of this fascinating technology, it is not surprising that companies and state funding agencies are now extremely reluctant to invest in new research and development projects for Stirling engines.

R&D at Frauscher Motors

Machines based on the alpha, beta and gamma principle in a power range from 0.5 to 11 kW were developed and manufactured in the course of our engine research. Test runs were carried out based on a test bench that was specially adapted to the requirements of Stirling technology and thanks to the availability of process data and analysis options. Optimisations in the areas of heat exchangers, phase angles, displacement distribution and regenerators were enabled through variable-speed performance tests. Manufacturing temperature-resistant and fatigue-proof heater-heat exchangers required considerable effort. We invested in a vacuum high-temperature furnace to carry out high-quality soldering and sintering processes to this end. Mastering the tribology in the area of the piston guides and piston rings, which must remain unlubricated in Stirling engines, proved particularly challenging. The development of permanently pressure-tight, helium-filled crankcase or generator housing demanded effort and patience. The use of hydrogen as a process gas is even more complex. Once a certain temperature is exceeded, this gas even diffuses through steel pipes. A selection of the manufactured and tested prototypes is shown in Figure 1.

Figure 1: Selection of Stirling engine prototypes from 2006-2016
Image: Prokop

Of the prototypes made, two engines with remarkable characteristics are worth highlighting. An oil-lubricated rhombic gear was developed for a Beta machine with a displacement of 250 cm^3. Heat was supplied using a commercial fuel oil burner, whose flue gas is directed axially into the interior of a rotationally symmetrical heat exchanger and subsequently flows out radially via the finely finned row of pipes, thereby transferring the heat. The piston rods are under atmospheric pressure, while the process chamber was filled with 50 bar helium.
The piston rods had to be sealed off from the process chamber in a way that prevented, on the one hand, the ingress of helium to the gear chamber, and, on the other hand, the ingress of lubricating oil to the process chamber. The fact that these sealing points pose an enormous challenge is clearly stated in several passages in the relevant literature. [1]

Even though the machine, shown in Figure 2, performed exceptionally well, in particular running extremely quietly and free from vibration, because of the aforementioned sealing problem, we were forced to surrender and come up with a new design to avoid such problem areas.

Figure 2: 250 cc Beta machine with rhombic drive

An extremely compact Alpha machine with a displacement of 600 cm^3 (Figure 3) proved to be our first notable success. The design of this machine avoided the weak points of previous designs, especially the sealing problem between the process chamber and the gear chamber. The solution was to fill the crankcase with the process gas – at the same medium pressure as that in the process chamber. This allowed us to additionally use the crankcase as a buffer space, which, acting on the undersides of the pistons, is indispensable in Stirling engines. The sealing separation point between the process chamber and the gear chamber/buffer chamber was thus moved to the piston rings, which make no claim to absolute tightness because the media and medium pressures are the same on both sides.

The machine, named A600, supplied a continuous electrical output of 5 kW, which was taken directly from the three-phase generator integrated into the crankcase. The machine was used in several external applications in research projects as a combined heat and power unit for wood pellets from 2013 to 2019. The heat was transferred via the flue gas in one case and via the fluidised bed process in two cases. The machine performed satisfactorily in all applications, although the operating times were only a few hundred hours in each instance.

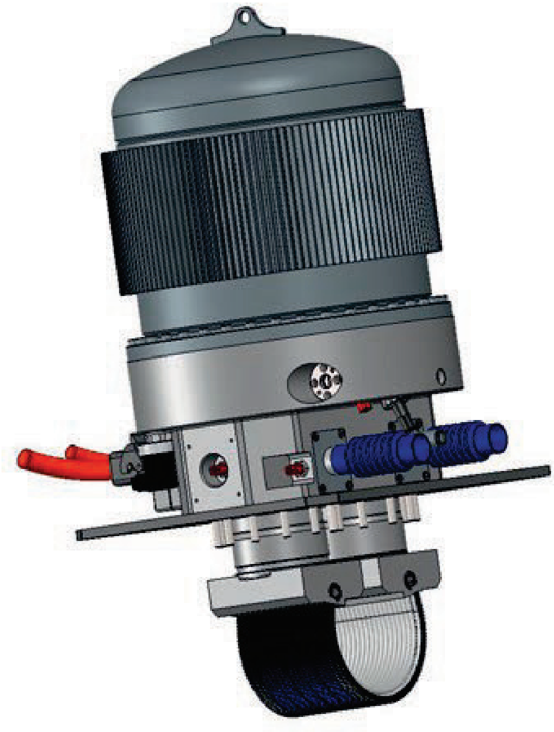

Figure 3: Type-A600 Stirling machine prototype

Image: Frauscher

Despite this machine's initially promising technology, deciding on series production and marketing eluded us for two reasons:

1) Even when limited to first-order balancing, the mass balancing of the two pistons moving parallel to each other and offset at a phase angle of approx. 130 degrees is extremely complex.

2) The invention of the alphagamma® technology led us to the finding that the piston forces can be halved at the same power output and, in addition, it is possible to achieve first-order mass balancing with only one counterweight on the crankshaft. The advantages of this technology are highlighted in detail in the following analysis.

Basic information on Alpha-Stirling machines

Developers of Stirling machines are certainly well advised to keep the design generally simple and robust and to use assemblies in the most multifunctional manner possible. For instance, it is a good idea to use the crankcase as a buffer chamber and generator housing at the same time. This type of design means the use of lubricating oil is prohibited, as its ingress into the process chamber via the piston rings cannot be permanently ruled out. The use of lifetime-lubricated rolling bearings is state of the art and, provided they are used well below their limit load, service lives of several tens of thousands of hours can be achieved. If the generator is arranged in the area of the crankcase, it is possible to avoid a pressure-tight and friction-intensive rotary transmission for the crankshaft to the outside and thus a sensitive wearing part of the machine. In a procedure which has its advantages, the generator may be used as the engine for starting the machine.

If the phase angle between the pistons can be selected at 90 degrees and both connecting rods are hinged to a crank pin, a considerable simplification is achieved. The fact that such a first-order mass balance can be attained using simple crank web counterweights at a position directly opposite the crank pin is common knowledge in the field.

Choosing a small connecting rod ratio, i.e. a relatively long connecting rod in relation to the crank radius, is also advantageous. As well as reducing the normal piston forces, this also renders second-order mass balancing unnecessary.

Without a doubt, one of the greatest challenges in terms of long service life is posed by the dry-running piston guides. In this regard, a balanced interaction between the surface hardness and roughness of the cylinder wall must be found using a suitable plastic compound on the piston skirt. Although the field of oil-free piston compressors provides some experience to draw from, the thermal and mechanical load conditions differ between compressors and Stirling machines. For example, the process gas helium or hydrogen behaves completely differently in the "lubrication gap" from ordinary air in compressors.

When considering dry-running guides, in the interest of minimising wear and lengthening the service life, the product of specific surface load multiplied by piston speed should obviously be kept as low as possible. Due to the compact design, the size of the running surface is limited. Therefore, in order to ensure a satisfactory maintenance-free operating time of the Stirling machine, measures to reduce the piston normal force must be taken. It should be noted here that, as well as favouring wear, a low piston normal force reduces frictional losses in the interests of a high overall efficiency.

Measures to reduce the piston normal force are:
→ minimal connecting rod ratio
→ process-related minimal piston force

The remainder of this article deals with the manner in which the piston forces can be reduced using a stepped expansion piston without affecting the net work per revolution.

The complex gas process

The working gas in a Stirling machine changes state in a highly complex manner. While an approximately isothermal course per section may be expected in the regenerator, the processes are quite polytropic in the cooler and heater heat exchangers and rather adiabatic in the cylinder chambers. An isothermal course in all sections is used as a basis in several calculation programs, which can lead to considerably erroneous results. Frauscher Motors therefore uses a polytropic simulation model, which constitutes an enhancement of the ideal adiabatic analysis.[1] It represents the actual conditions in the machine very closely, which significantly increases the calculation accuracy. The p-V diagram in Figure 1 shows the differences between the results of the isothermal, adiabatic and polytropic calculation methods compared with those of the measured values of an executed machine on the test bench.

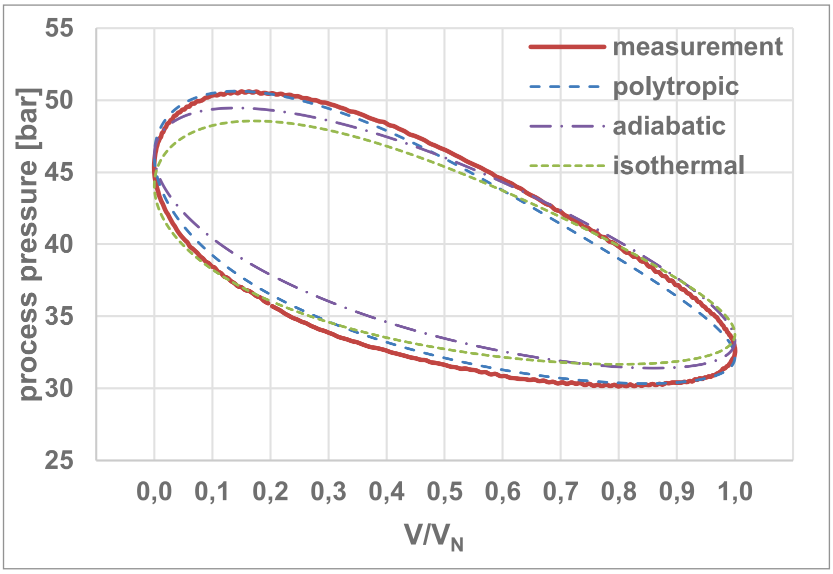

Figure 1: Comparison of the calculated and the measured p-V process curve of a Stirling machine

The polytropic calculation method, which experience has shown to be quite accurate compared to practical applications, is used in the further explanations.

Task

The following calculations are carried out based on an Alpha machine with a fixed expansion displacement, fixed heat exchanger volume and variable compression volume and phase angle, with the net work per revolution being kept at the same level in each variant by adjusting the process medium pressure. To obtain a direct comparison with previously executed alpha machines, the calculations are extended to a variant with stepped pistons (alphagamma® process). A Tk/Th temperature ratio of 300K/900K is assumed for all variants. Furthermore, the buffer volume (gear chamber) is assumed to be sufficiently large in terms of negligible pressure oscillations.

The results should show the absolute value of the work of the individual pistons per revolution, the extent of the total positive and negative work of both pistons and the piston forces that occur in the individual variants. The examined Alpha machine with the fixed and variable variables is illustrated in Figure 2.

The data of the selected variants and their displacement and phase angle is listed in Table 1.
A vectorial representation of the expansion volume, the two compression volumes and the size and phase angle of the resulting compression volume for the alphagamma® process is shown in Figure 3.

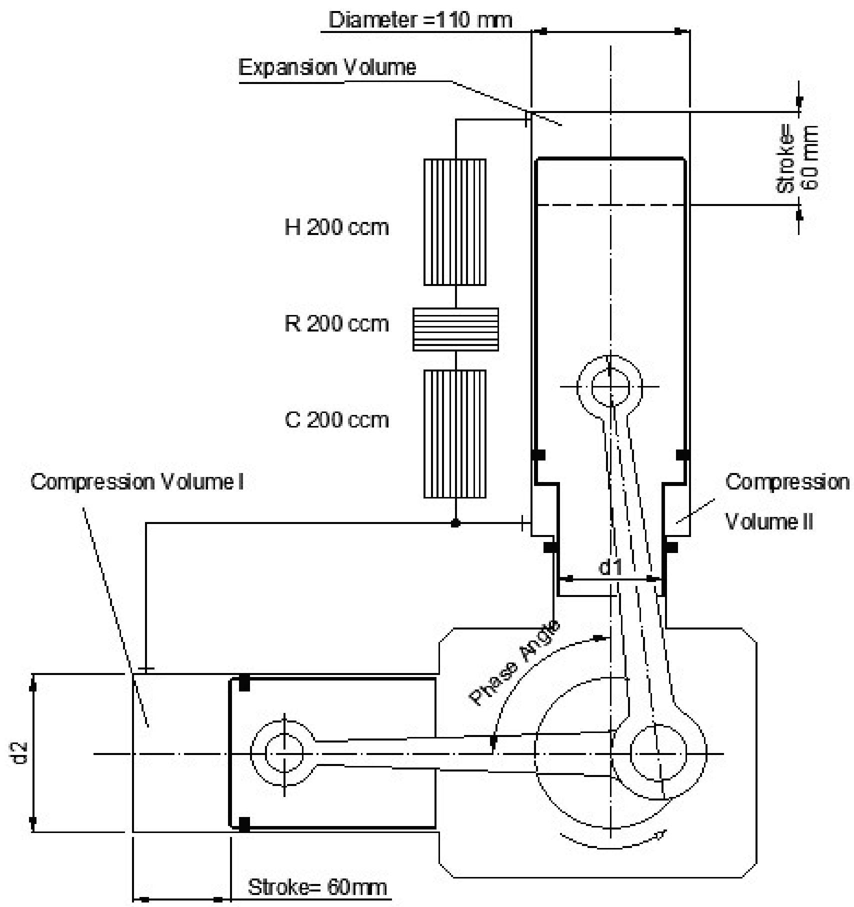

Figure 2: Illustration of the alpha machine and its fixed and variable dimensions

Variant a) standard alpha	Values		Comment
Compression volume (d2= Ø 110 mm)	570	cm³	
Connecting rod length Compression piston	230	mm	variant b, c also
Compression piston mass	4,5	kg	variant b, c also
Expansion volume (d1=Ø 110 mm)	570	cm³	variant b, c also
Connecting rod length Expansion piston	230	mm	variant b, c also
Expansion piston mass	4,5	kg	variant b, c also
Phase angle	90	Degree	
Rotational speed	1.000	U/min	variant b, c also
Temperature ratio	300/900	K	variant b, c also
Designation d1/d2 angle			110/110-90°
Variant b) optimized alpha			
Compression volume (d2= Ø 90 mm)	382	cm³	
Expansion volume (d1= Ø 110 mm)	570	cm³	
Phase angle	135	Degree	
Designation d1/d2 angle			110/90-135°
Variant c) alphagamma®			
Compression volume (d2 = Ø 80 mm)	302	cm³	
Ring volume (d1 = Ø 75 mm)	305	cm³	
Compression volume resulting	429	cm³	
Expansion volume	570	cm³	
Resulting phase angle	135,3	Degree	
Designation d1/d2 angle			75/80-135°

Table 1: Summary of volume and angle variations

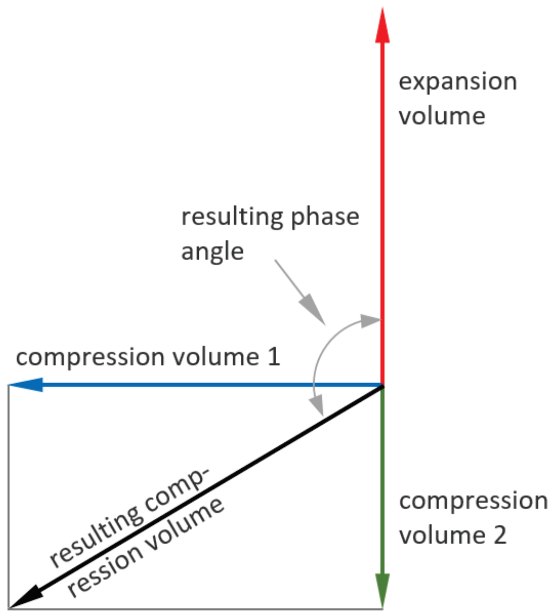

Figure 3: Vectorial representation of the phase angle and the resulting compression volume in the alphagamma® process

Description: (see Figs. 2, 3, 4, Table 1)

Variant a) Standard alpha process

This is a “traditional” alpha configuration with a v-shaped cylinder arrangement, 90 degree cylinder angle (= phase angle) and equal displacement in both cylinders. Since the small diameter and the large diameter (110 mm) of the stepped piston are the same, no annular volume is present. This variant thus corresponds to a standard alpha configuration, which can often be found in publications and in the technical literature. It has also been executed in practical applications and placed on the market in several cases.

An observation of the indexed work over one crankshaft revolution determined based on the polytropic calculation method, shown in Figure 4, indicates high negative work shares. Compensation for these through even higher positive sections is required in order to obtain the desired net work. As the following calculations show, a progress of work like this generates extremely high pressure strokes, accompanied by the highest piston forces. These forces are also reflected in the torque curve subsequently. A “round” run therefore requires corresponding flywheel masses to ensure that the current fluctuations in the generator current remain within the permissible value range.

Variant b) optimised alpha process

The insight that a design based on variant a) leads to unsatisfactory results was revealed by the research activities of Frauscher Thermal Motors at an early stage. A series of tests carried out over the course of several years showed the following. The negative work shares per revolution can be significantly reduced using phase angles between 120 and 140 degrees as well as a compression volume amounting to only 65–75% of the expansion volume. Starting in 2012, Frauscher Motors developed a machine with 5 kW electrical power, type designation A600. This was manufactured in several versions and successfully operated in internal and external applications. The progress of work – shown in figure 4 – clearly illustrates that the negative work shares drop to less than 10% in relation to the positive work.

However, the complicated mass balancing, which could no longer be carried out using simple counterweights for the aforementioned phase angle range, proved time-consuming. The search for simpler designs – in addition to the striving for longer service lives – finally led to the alphagamma® solution.

Variant c) alphagamma® process

This process uses the stepped piston. As in variant a), the compression piston is in a phase offset of 90 degrees (in the sense of a simple mass balance). The compression volume is divided into two compression chambers. The annular volume (compression volume II) has a phase shift of 180 degrees to the expansion volume. Taking account of the compression volume of the compression cylinder (compression volume I), which is phase-shifted by 90 degrees, it forms a resulting phase angle of 135.3 degrees. The resulting phase angle is calculated in a simple manner using the angle functions as shown in figure 3. The length of the arrows is in direct proportion to the corresponding displacement.

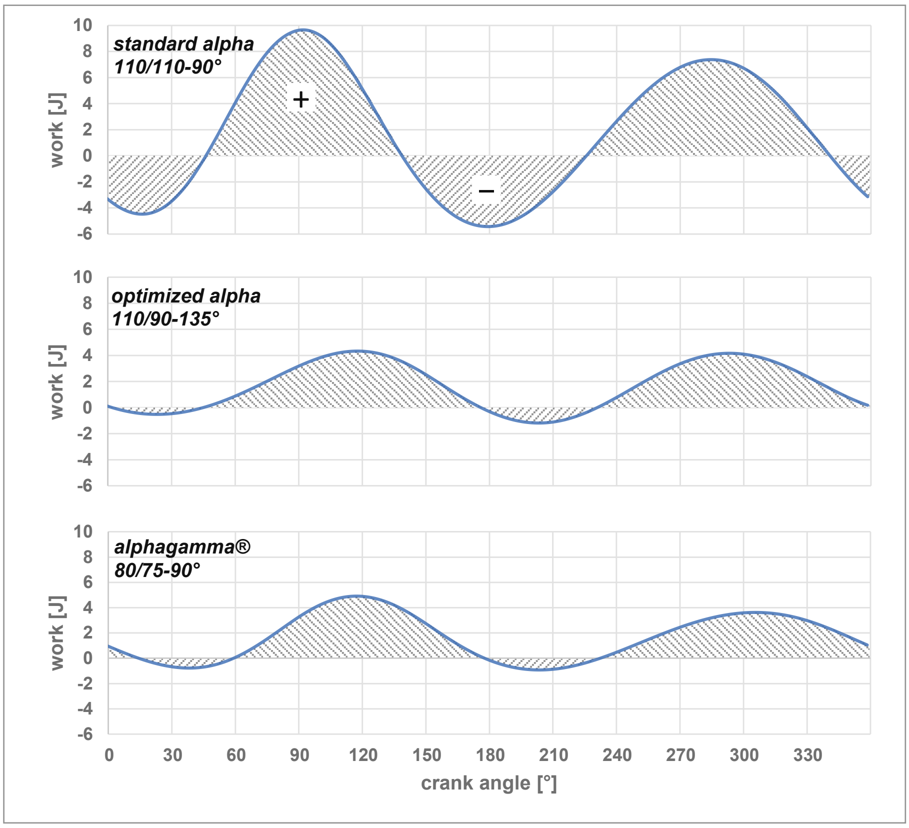

Figure 4: Results of the positive and negative work shares per cycle

Looking at the work histories in Figure 4 and comparing the optimised alpha process with the alphagamma® process, no advantages over simple mass balancing are initially apparent. The work values listed in Table 2 confirm this in absolute figures.

	standard alpha 110/110-90°	optimized alpha 110/90-135°	alphagamma® 75/80-90°
positive work [J]	1081,8	648,6	647, 7
negative work [J]	-494,6	-57,9	-55,3
Ratio	45,7%	8,9%	8,5%

Table 2: Sum of positive and negative work shares per cycleThe effect of the stepped piston

The effect of the stepped piston

The additional advantages of the alphagamma® process only become apparent when the piston work and the piston forces are considered separately. Figure 5 illustrates that the expansion piston basically has to do the negative work of the compression piston in the alpha process. Only the work of the expansion piston that goes beyond this can be used as net work. In the alphagamma® process, on the other hand, the compression piston even contributes a small positive amount to the net work delivered, significantly lightening the load of the expansion piston.

Figure 5: Distribution of work among the pistons

The calculation results of the piston forces show the individual work of the pistons. The maximum gas force on the pistons is shown in Figure 6. This takes account of the fact that the gas pressure in the buffer space acts on the underside of the pistons and thus the piston forces were calculated from the difference between process pressure and (constant) pressure in the buffer space. Furthermore, our calculation method also incorporates the losses in pressure via the heat exchangers – based on a gas exchange frequency at 1,000 rpm.

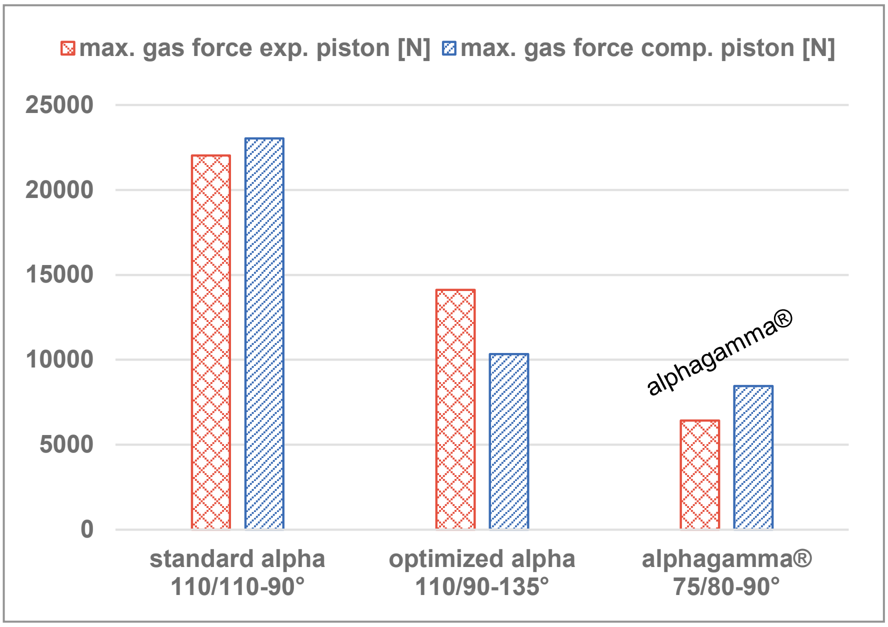

Figure: 6: Maximum gas differential forces on the pistons

The actual piston force acting on the crank mechanism can only be represented by including the mass force. This reveals another advantage of the alphagamma® method: Assuming a tried and tested piston mass of 4.5 kg for each of the two pistons and a speed of 1,000 rpm, the mass force hardly has any increasing effect on the maximum piston forces. In the expansion piston, the gas force dominates in any case, reaching its maximum value at around mid-stroke. In the case of the compression piston, the mass force counteracts the piston forces resulting from the differential pressures at the reversal points, which leads to a significant reduction in the maximum piston force, as shown in Figure 7.

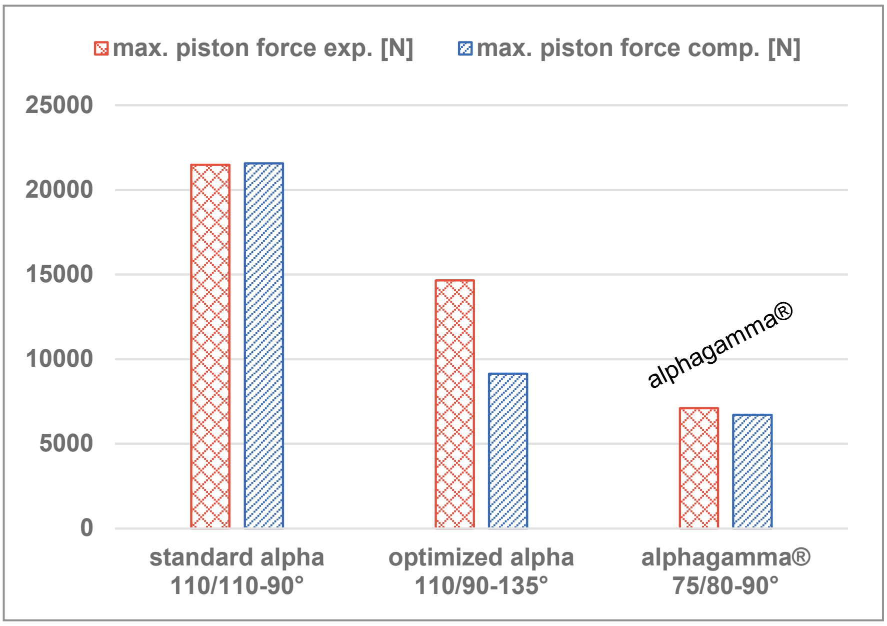

Figure 7: Piston forces taking account of the mass force

The calculation results of the piston forces and the piston work are summarised in Table 3. The mean process pressures are set so that the total work of each variant reaches an equal level.

	standard alpha 110/110-90°	standard alpha 110/110-90°	aphagamma® 75/80-90°
mean process-pressure [bar]	33	54,5	50,5
max. piston force compression [N]	21573	9142	6718
max. piston force expansion [N]	21482	14650	7100
work compression piston [J]	-863,6	-642,7	87,7
work expansion piston [J]	1454,2	1233,2	503,4
total work [J]	590,6	590,5	591,1

Table 3: Comparison of the variants considered in table layout

It is shown that the use of a stepped piston as an expansion piston with suitable dimensioning drastically reduces the negative work per cycle, and therefore also the compression loads on the machine, as well as the individual work of the pistons and their maximum forces. In a design example with an expansion volume of 570 ccm and the same compression volume and a phase angle of 90 degrees (standard alpha process), the maximum piston force of the expansion piston is three times higher than in the alphagamma® process including a piston mass used in practical applications. For the compression piston, the piston force is as much as 3.2 times higher, as shown in Table 3.

In the alphagamma® process, this paves the way for lubrication-free dry-running technology of the pistons due to the low piston normal forces and, overall, a simple, inexpensive and reliable machine design for long maintenance intervals. As well as being based on theoretical calculations, the data of the variants designated as “standard alpha”, “optimised alpha” and “alphagamma®” can be verified by test bench results in executed machines.

There’s more...

The friction losses of the piston guides due to the piston normal forces as well as the losses of the piston ring and bearing friction were not taken into account in the previous explanations. The focus was on the acting gas differential pressure forces and the resulting work of the pistons and, in addition, on the effects of the mass force.

The results of the previous analyses are suitable for a closer look at the friction of the piston guides.

The friction work of the piston guides of both pistons per crankshaft revolution is shown in Figure 8 below. We assume that the piston skirts are coated with a proven plastic compound and that the occurring normal forces are determined based on the course of the process pressure and the crank angle with the known connecting rod ratio of crank radius/connecting rod length = 0.1304. An average value of 0.2 is assumed as the coefficient of friction.

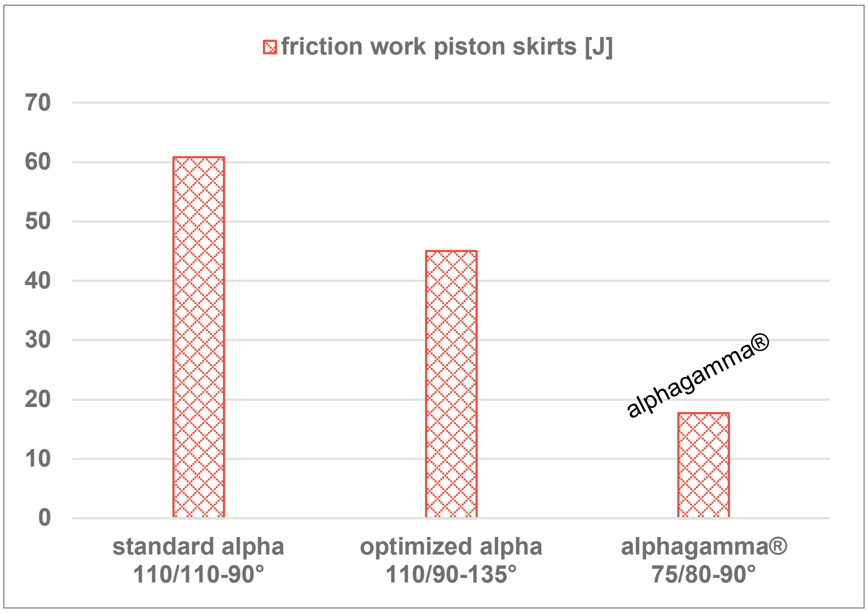

Fig.: 8 Piston – friction work per cycle

The consequences of high differential pressure forces and piston forces could unquestionably be extended to the friction of the piston rings and the crankshaft and connecting rod bearings also. This is omitted here as experience has shown that the effects at these points are manageable. In this respect, we merely point out that “moderately” loaded rolling bearings can achieve service life values that are extremely satisfactory in terms of maintenance-friendly operation, even when lubrication is lacking.

Rather, we recall the task mentioned at the outset, i.e. that the net work per revolution should be equalised to a consistent value for all variants by adjusting the process medium pressure. Since the friction work reduces the usable net work, in order to make a reliable comparison, we want to increase the mean process pressure for each variant by the value that leads back to the total work of 591 joules shown in Table 3. Finally, the actual piston forces that occur with the same net work and additional inclusion of the friction work on the pistons are shown in Figure 9 and Table 4.

	standard alpha 110/110-90°	optimized alpha 110/90-135°	alphagamma® 75/80-90°
mean process-pressure [bar]	36,4	59	52
max. piston force comp. [N]	24235	10070	6947
max. piston force exp. [N]	23225	15438	7119
total indicated work [J]	652	636,6	609,5
friction work piston skirts [J]	60,8	45,0	17,7
net shaft work [J]	**591,2**	**591,6**	**591,8**

Table 4: Comparison after pressure correction

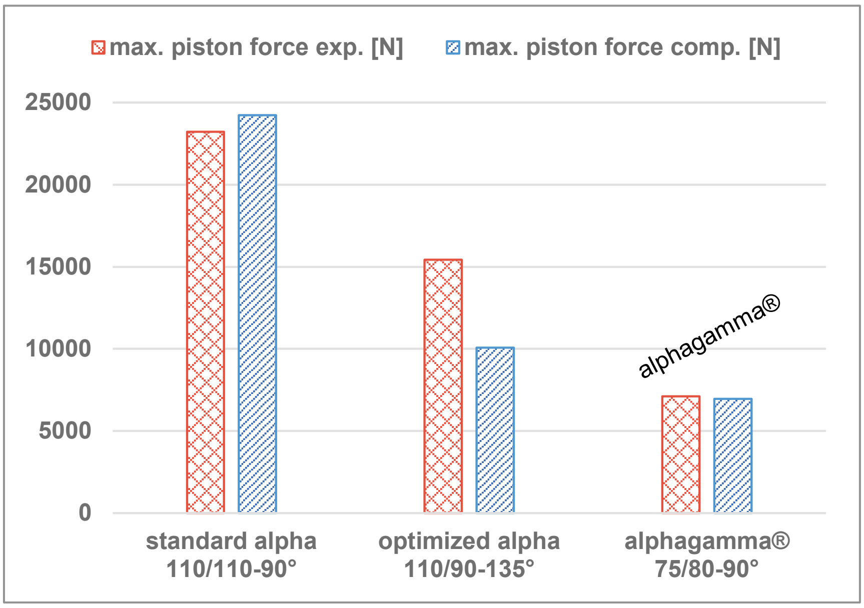

Figure 9 : Actual piston forces including friction forces

As was to be expected, the piston forces in variants a) and b) continue to increase as a result of the improvement in the process medium pressure. In the alphagamma® process, on the other hand, the values between the two pistons move in such a way that the compression piston force increases somewhat, while the expansion piston force remains practically the same. The differences between the standard alpha process and the alphagamma® variant increase to the factors 3.5 (compression piston) and 3.3 (expansion piston). If the effects of piston ring friction and bearing friction were to be included, the factors would be even somewhat higher.

Torque curve: Reflection of the process work

Fig. 10 shows the torque curve of the considered variants at the crankshaft based on the piston forces in accordance with Table 4 and Figure 9. The representation reveals the rather soft, sinusoidal torque curves that are standard in Stirling engines. As expected, high negative shares occur in variant a), which lead to an unsteady operation affected by torsional vibrations. The negative shares hardly occur at all in variant c). Instead, two distinctly positive, sinusoidal torque surges are emitted. These result in refined and quiet running, the uniformity of which can be achieved even with a relatively small flywheel mass. This can be designed as a rotor of an electric machine in the sense of high integrity.

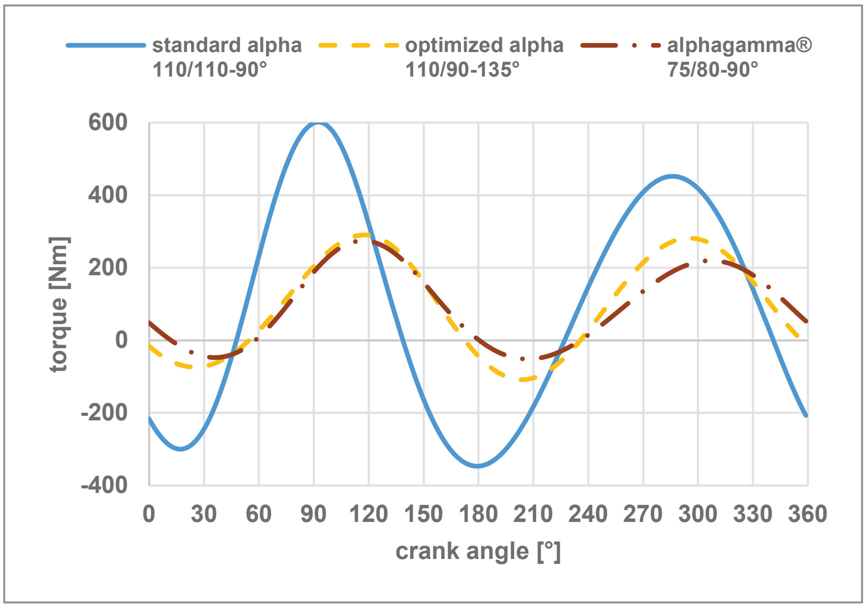

Figure 10: Torque curve of the variants based on the piston forces in accordance with Figure 9

Summary

The effects on the piston gas forces, piston friction and torque curve are presented and compared in each case with a stepped piston based on the alphagamma® method. The presentation is based on a Stirling Alpha configuration with different compression volumes and phase angles.

It transpires that the piston forces of a machine in accordance with the alphagamma® method are less than 30% of those in accordance with the often published method, called “Standard Alpha” in this case. Equal work per cycle was taken into account. The consequences are essential for the performance of the new engine technology.

Firstly, the low piston forces allow operation without lubricating oil. In a procedure which has its advantages, the connecting rods are linked directly to the pistons. Thus, the normal piston forces can be relieved without the need for crosshead guides or similar structures. This is carried out using dry-running plastic guides mounted on the piston skirts.

While the friction work of the observed sliding pairing in the standard Alpha process amounts to almost 10% of the useful work, in the alphagamma® process, this is reduced to less than 3%. Any further consideration of whether and how additional mechanisms must be integrated to relieve the piston normal forces is therefore unnecessary. The low friction has a significant influence on the considerable overall efficiency, as shown in the example of a 6 kW class alphagamma® machine based on the test bench measurement data. (Figure 11)

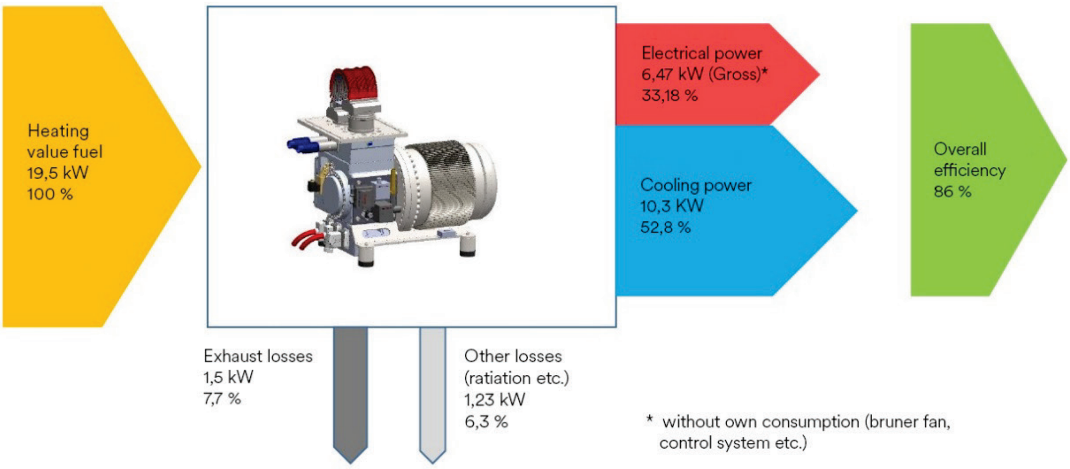

Figure 11: Energy flows and overall efficiency of the G600 Stirling machine

The lubricant-free gearbox allows the crankcase to be used as both a process gas buffer chamber and generator housing. The number of moving parts in the machine is reduced to a minimum: Two pistons, two connecting rods and one crankshaft (Figure 12).

Figure 12: Ratio of the number of moving parts in a petrol engine (16) to those in a alphagamma Stirling engine (5)

Image: Frauscher

Finally, the effects of the comparatively low piston forces on the overall weight of the machine must not go unnoticed. If consistent lightweight construction is applied, not only the crank mechanism, but also the gearbox housing, can be significantly streamlined. Since the operating noise is low for design-related reasons, the expense of sound insulation is also avoided. Thus, the overall weight of the system can compete with engines with internal combustion in many cases.

More information:

Frauscher Motors GmbH
Gewerbestraße 7 | 4774 St. Marienkirchen | AUSTRIA
t: +43 7711 3182

www. frauscher-motors.com

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