The steam hybrid engine
(Hansen engine)

- Function -
- Optimisation -
- Power -
- Efficiency -

using the example of a boat diesel engine

- PROPOSAL -
for a dissertation topic

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1 Foreword

The simple design and technical maturity still make combustion engines attractive in the present. Added to this are the simple raw materials they are made of, in contrast to modern machines that are being developed or are already in use as part of the fight against climate change. A true comparison between electric machines and combustion engines, however, is hardly possible. It is true that electric machines are also energy converters, but from electrical energy to mechanical energy. Electrical energy, however, does not occur in nature in usable form and must first be generated artificially from naturally occurring raw materials. Combustion engines, especially heavy marine diesel engines, are capable of this. They convert naturally occurring raw materials such as heavy oil into mechanical energy, from which electricity can then be generated almost without loss using an electrical machine, the generator. Modern technologies then store this power, with losses, in accumulators, which in turn convert their power into mechanical energy in another electric machine, the electric motor, also with almost no losses.
These are major detours, and there is a lack of compact plants to generate electrical energy directly from nature's raw materials. Wind power plants and photovoltaics therefore already largely determine the appearance of formerly natural areas, to the annoyance of local residents. However, the efficiency of these plants can hardly be surpassed. Ultimately, however, the conversion of energy into mechanical energy is once again of decisive importance. The production of hydrogen by means of the electrical energy directly obtained with it again has a poor efficiency, and the conversion of hydrogen into mechanical energy also has a poor efficiency, so that overall a comparison with internal combustion engines becomes possible, because the efficiency of internal combustion engines is also poor. After all, the most sophisticated models of marine diesel engines already achieve an efficiency of 50 %. But if you consider this, you realise that just as much energy is released in waste heat as in mechanical energy. In fact, the ship always has to bunker fuel for twice the distance. This is unacceptable when, for example, cars have to refuel for three times the distance. If they drive with petrol engines, it is even four times the distance. But with a hydrogen fuel cell vehicle, the calculation is hardly any better.
The author has now made it his task to focus on the larger part of the energy occurring during energy conversion and to investigate its usability - the use of waste heat from combustion engines. This is not a new topic. The literature shows various inventions on the subject. In most cases, the realisation fails due to the working medium water, whose boiling point must first be reached before it becomes a compressible medium, namely steam, with which work, mostly in the form of expansion, becomes possible. One example is the vehicle supplier Mahle, which offers waste heat recovery units for vehicles, but these units are not used to supplement the main drive, but only to operate the air conditioning or similar auxiliary units or to feed an accumulator. At best, the efficiency is improved by 5 percentage points. The waste heat recovery systems, however, are currently the best the market has to offer in terms of waste heat utilisation.

This paper will now focus on ship propulsion, as experts and industry insiders believe that the diesel engine will continue to dominate shipping for decades to come. Global greenhouse gas emissions from this essential part of the transport industry are 3% - a figure not to be underestimated - and despite great efforts, no alternative is yet in sight. It is simply the huge distances involved in global trade. And range is always limited by the installation space of the vehicle if it is to be self-sufficient. After all, it is supposed to have cargo capacity above all. This problem is reminiscent of space travel. For the moon landing, a rocket was needed that consisted of more than 99 percent by weight of propulsion and fuel. Yet it still had to be multistage to arrive and also make the return journey. An

earthly example of solving the transport problem is the new Silk Road, a trade route that connects China with Europe by rail. In theory, electrification would also be conceivable here, since it runs over land. But here, too, the diesel engine is used. Furthermore, the diesel engine still forms the backbone of commercial vehicles, which are also part of the merchant navy and dominate the transport problem. Here, too, research is being conducted into alternatives, but the principle of range and installation space applies here as well. One also speaks of power density. The diesel engine has mastered this for over a hundred years. Although hydrogen is always in mind, it is climate-neutral if it is produced with wind power or photovoltaics, but it always throws a spanner in the researchers' works when it comes to storing, storing and transporting it.
Should this problem be solved one day, the time of the combustion engine is far from over, because hydrogen burns and thus makes expansion work possible as it is the basis of the working principle of combustion engines. The company Keyou from Munich is already converting city buses with diesel engines into hydrogen combustion engines. The advantage over the hydrogen fuel cell, which provides electrical energy, is the saving of valuable resources. Internal combustion engines are undemanding. Electric drives require multi-stage energy conversions, which makes them complicated and uneconomical compared to the internal combustion engine.

2 Introduction

This is the background to the Hansen engine. Although it is only one of many inventions for waste heat utilisation in combustion engines, it is one thing above all: simple. Simplicity is associated with economy and robustness, should it work. The inventor Prof. h.c. Uwe Hansen from Albersdorf in Schleswig-Holstein, at any rate, could not demonstrably justify this. His efforts were mainly aimed at selling the patent. His practical experiments did not enable any authoritative measurements. Moreover, they were not based on any theoretical foundation, which is to be made up for with this investigation. Even the television report on Germany's regional television showed the reason for his failure in the first few seconds: Hansen's theoretical basis is the steam engine, whose efficiency is at best not even half that of the diesel engine, and that is why it no longer exists in the present day. Every engineer who has to judge the patent knows that.
However, Hansen's steam engine has one fundamental feature: the working medium. There has never been a steam engine that did not work with water. Hansen proposed butane, which is, however, flammable and can escape via the main leakage point, namely the piston ring. The decision-makers will always have stumbled over this passage at the latest.
These two construction details will also have to be taken into account in the investigation. First, however, the machine description in the form of an expert opinion by Dr. Sönke Harm, University of Kiel, dated 21.05.2005 (translation):

"The so-called "Hansen engine" in its present form is a modified 4-stroke petrol engine consisting of three cylinders, which has been supplemented by a single-cylinder heat engine running in quasi 2-stroke mode, which delivers its work directly to the same crankshaft. The energy required to operate the heat engine is taken from the exhaust gas of the three cylinders running in Otto mode on the one hand and also from the engine's cooling circuit on the other. The aim of this system is to achieve a significant increase in overall efficiency, which has already been demonstrated in initial tests.
Butane is used as the working medium for the heat engine running in a closed cycle due to its boiling temperature. The butane is pumped in a liquid state from a reservoir by means of roller cell pumps at a pressure of approx. 25 bar into a heat exchanger with good efficiency. There, the evaporating butane absorbs the heat energy of the exhaust gas flow of the three combustion cylinders working according to the Otto principle. On the fourth cylinder, the timing of the intake and exhaust valves is altered by a modified camshaft in such a way that the butane gas, which is at relatively high pressure behind the heat exchanger, enters the cylinder at every (and not just every second) crankshaft revolution and, by expanding up to the hermetically sealed piston, delivers its thermal energy directly to the crankshaft in the form of mechanical work. During the subsequent stroke movement of the piston, the expanded butane gas is conveyed through the exhaust valve into a cooler, where it is liquefied again and flows back into the reservoir.
The overall concept is convincing due to a comparatively low conversion effort of a conventional Otto engine in combination with an already quite good technical state of development. Since the overall efficiency of a petrol engine in particular is quite low at approx. 25 to 30 [%] due to the relatively high energy losses in the exhaust gas and in the cooling circuit, energy recovery from these areas in principle holds quite a high savings potential. In terms of mechanical construction, series production of an engine based on the principle presented seems quite realistic in terms of production costs and durability. Further adjustments could then be made, such as modifying the crankshaft for a uniform ignition distance. Because of the probably less rapid controllability of the power output compared to a petrol engine (slow-running heat transfer processes), an application for

predominantly stationary-running engines, e.g. in power generation, would initially suggest itself. Beforehand, however, the increase in efficiency actually achieved must be demonstrated metrologically on an engine test bench over a sufficiently long period of time in order to be able to quantify the realisable reduction in consumption and the associated savings in fuel costs, which can be used as a basis for assessing the economic viability of series production. In addition, detailed measurements of a number of operating parameters are required in order to verify the correct dimensioning of the individual components metrologically."

The stationary running of power generators is also found in ship propulsion systems. The practical implementation of the machine in the form of a prototype, which is still pending, may show whether commercial vehicles in the form of long-distance trucks or even rail vehicles also fit into this category for the Hansen engine.
To be clear, the thermal efficiency of the steam part of the machine examined in the following is 0.12. With an efficiency of modern large diesel engines such as those installed in container cargo ships, it is 0.50. The overall efficiency of the Hansen engine refers to the efficiency of the diesel engine and the efficiency of the steam part. Since the waste heat of the diesel part is only 50 % here, the share of the steam part in the overall technical efficiency is reduced to 6 %. This is reminiscent of Mahle's waste-heat-recovery unit, whereby friction losses are not even taken into account. However, in view of the high pressure of the working medium of up to 35 bar that is likely to be available and which acts on the piston crown, this value seems downright ridiculous.

In this study, therefore, a theoretical alternative to the calculation of the technical efficiency compared to the thermal efficiency is to be offered in detail by means of harmoniously coordinated efficiency of the machine components in an engineering-mechanical way. This can then be helpful in the evaluation of a test in the form of a metrological examination of a prototype, should Prof. Hansen be correct in his assessment of a large increase in overall efficiency.

3 Function

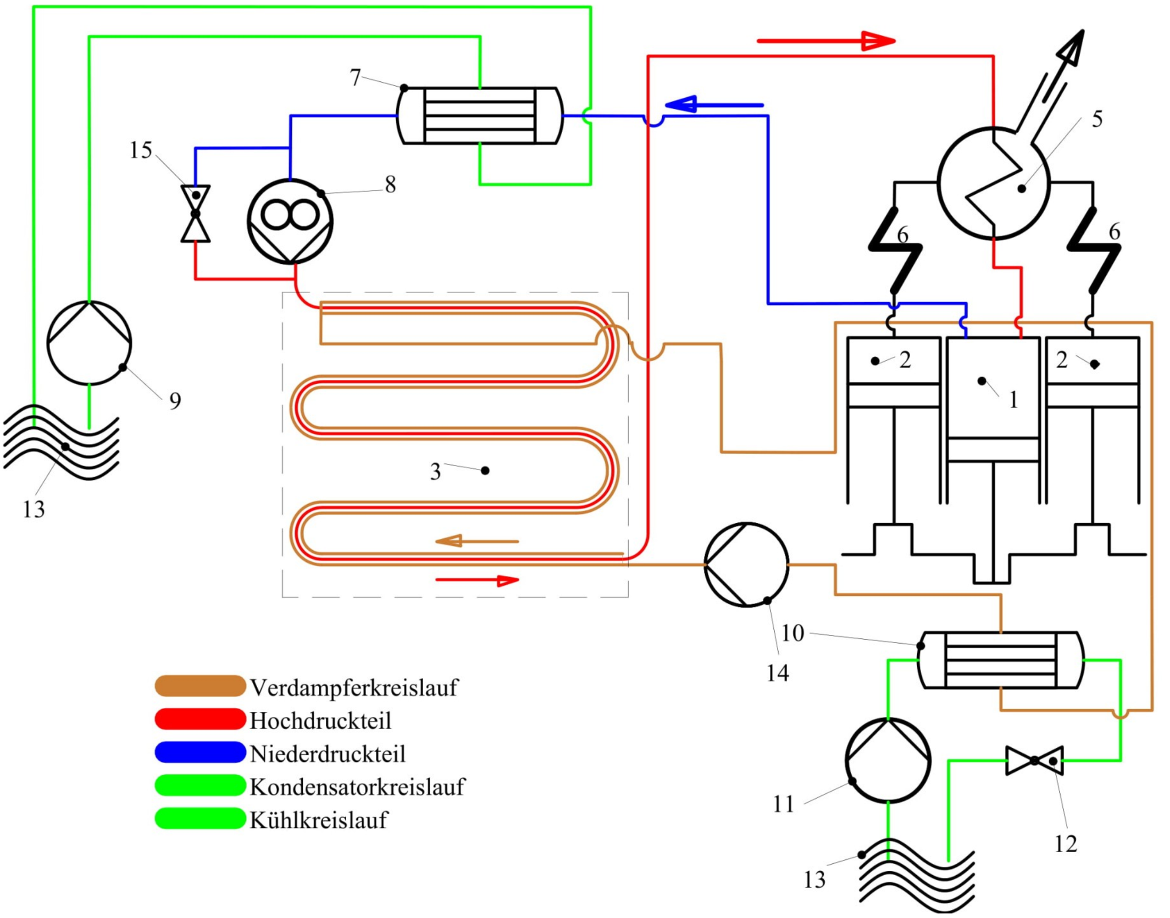


Figure 1: Function plan

3.1 Functional description

3.1.1 Evaporator circuit

The evaporator circuit is used to heat and evaporate the n-butane in the tank (4). The combustion of the diesel fuel in the cylinders (2) generates heat which is transferred to the fresh water cooling circuit of the engine (2, 1, 2). The water pump (14) delivers the heated fresh water into the double-tube evaporator (3). The tube bundle in which the hot fresh water flows in the opposite direction to the working medium heats it so that it evaporates. . A seawater heat exchanger (10) is also part of this circuit (see 3.1.5).

3.1.2 High pressure part

The working medium pressurised in the double-tube evaporator (3) presses into the superheater (5) o. Hansen heat exchanger, whose tube bundle, in which the working medium flows, is heated by the exhaust gas of the combustion engine cylinders (2) (6), so that the working medium is heated to ~130 °C according to the inventor Prof. h.c. Hansen. Hansen to $p_1 \approx 26$ bar pressure if it is n-butane The working medium then expands in the steam cylinder (1).

3.1.3 Low-pressure part

Then, with the outlet valve open and the steam inlet valve closed, the piston pushes the steam out of the steam cylinder (1). From here, the vapour, which is now under ~9 bar and ~85 °C, flows into the condenser, where it cools down and liquefies again. The downstream feed pump (8) pumps the liquefied gas back into the tank.

3.1.3.1 Bypass

In order to stop the engine quickly, the pump has a bypass which can be opened with the valve (15) so that the high and low pressure sections can equalise their pressure and the steam cylinder (1) can no longer deliver mechanical work.

3.1.4 Condenser circuit

The sea water pump (9) takes water from the sea (13) to cool or liquefy the working medium in the low-pressure section, which flows through a tube bundle in the condenser (7). After the seawater has flowed through the condenser, it flows back into the sea.

3.1.5 Cooling circuit

In warm regions of the seas, the engine may overheat because the heat output of the evaporator (3) in the tank (4) to the working medium for cooling the fresh water is not sufficient to keep the engine at operating temperature. The dual-circuit cooling system provides a remedy for this case. The shut-off valve (12) or *sea valve* is opened. The pump (11) sucks in sea water (13) and leads it into the heat exchanger (10) where it flows around the pipe bundle that leads the fresh water from the evaporator circuit (see chapter 3.1.1).

4 Vapour pressure curve of the working medium

The sealing of the piston against the cylinder of the steam part is the critical point of the Hansen engine. Therefore, n-butane must not be used. The risk of fire is too high. Therefore, a working medium called R1234ze(Z), which is originally used in high-temperature heat pumps, was chosen as an alternative. It is flame retardant and non-toxic [1]should it be released into the environment. Its performance is similar to n-butane:

Critical temperature	T_c	423.27 K	[2]
Critical pressure	p_c	35.30619 bar	[2]
Boiling point	T_{sp}	---	
Boiling pressure	p_{sp}	1.01325 bar	
Temperature at the triple point	T_{Tri}	273.0 K	[2]
Pressure at the triple point	p_{Tri}	0.67849 bar	[2]

The vapour pressure curve must be determined from the proportionality of the Clausius-Clapeyron equation as follows: It conceals a linearity of the logarithmised vapour pressure in bar over the reciprocal temperature in K [3]: $y = m \cdot x + b$.

$m = \frac{\ln(p_2) - \ln(p_1)}{\frac{1}{T_2} - \frac{1}{T_1}}$ and $b = \ln(p) - m \cdot \frac{1}{T}$. With $p_2 = p_c$, $p_1 = p_{Tri}$, $T_2 = T_c$ and $T_1 = T_{Tri}$ the vapour pressure equation becomes:

$$p(T) = e^{\frac{m}{T} + b} \quad (1)$$

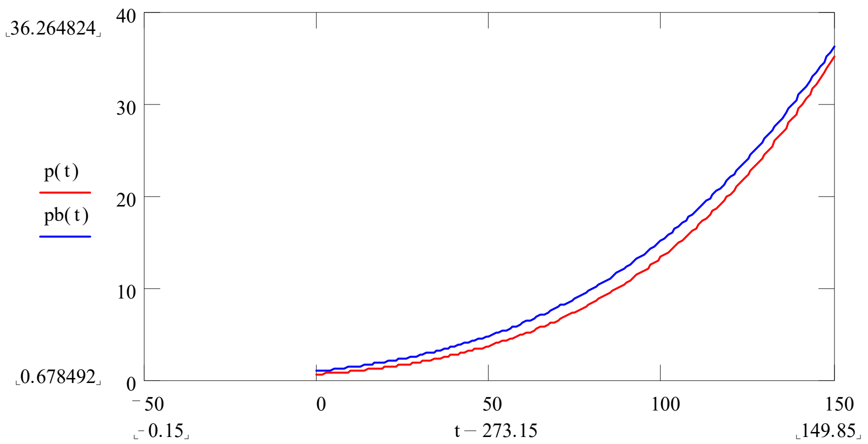


Figure 2: Vapour pressure equations n-butane

The abscissa represents the temperature in °C and the pressure of the working medium in bar on the ordinate. As with n-butane (blue), R1234ze(Z) (red) also shows a strong pressure increase in the range of 50...150 °C, as was already ideal for cooling water (~100 °C) and exhaust gas (~1000 °C) with n-butane as the working medium for use in engine construction.

4.1 Polytropic expansion

In thermodynamics, n-butane is treated as an ideal gas, so that for the adiabatic expansion a polytropic expansion with $p \cdot V^{\kappa} = const$. [4]is assumed and thus becomes easy to calculate. The polytropic exponent κ is 1.13 for n-butane. Based on the similarity of the vapour pressure curves, similar things can be assumed for R1234ze(Z), and κ will not differ significantly, but must be determined:

From the steam pressure equation for R1234ze(Z) a pressure of p_1=24.67 bar results for the temperature of T_1=130 °C. The following applies for the adiabatic expansion $\frac{p_1 \cdot V_1}{T_1} = \frac{p_2 \cdot V_2}{T_2}$ [5]

The volumes are now the control variables of the expansion and can be assumed. From the steam engine, however, we know that with $\frac{V_1}{V_2} \approx \frac{3}{10}$ [6]was used. T2 is sought. The pressure after the expansion p_2 can be replaced with the steam pressure equation $e^{\frac{m}{T_2}+b}$ can be substituted.

Unfortunately, a conversion of the equation $\frac{p_1 \cdot V_1}{T_1} = \frac{e^{\frac{m}{T_2}+b} \cdot V_2}{T_2}$ to T_2 is not possible, and it must be iterated:

$$
t2 := \left|
\begin{array}{l}
t2 \leftarrow 273.15 + 50 \\
\text{while } \frac{p1 \cdot 3}{t1} \geq \frac{e^{\frac{m}{t2}+b} \cdot 10}{t2} \\
\quad t2 \leftarrow t2 + 0.00001 \\
t2 \leftarrow t2
\end{array}
\right.
$$

With the values for p_1, T_1, V_1 and V_2, this results in a T_2 of 67.96 °C. From the vapour pressure equation it follows that p_2 = 6.26 bar.

The polytropic expansion provides $p_1 \cdot V_1^{\kappa} = p_2 \cdot V_2^{\kappa}$. From this it is determined $\kappa = \frac{\ln\left(\frac{p_2}{p_1}\right)}{\ln\left(\frac{3}{10}\right)}$ to 1.14.

This result is successfully checked with the further equation for the adiabatic expansion after the temperature with $T \cdot V^{\kappa-1} = const$. is successfully checked.

5 Example machine

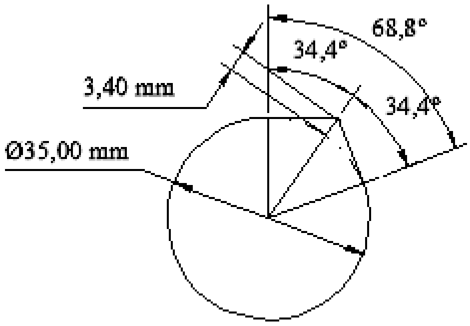
The YANMAR 3 GM 30 F is favoured as the basis for the conversion. It is a common boat engine with dual-circuit cooling as required in the function plan. There are many used examples of this engine. Performance and consumption diagrams follow in chapter 9

Cylinder:	3	[7]
Bore x stroke:	75 x 72 mm	[7]
Displacement:	0.954 l	[7]
Performance:	20.3 KW at 3600 min-1	[7]

6 Constructive details

- For the conversion of an existing engine, such as the example machine here, it is necessary to use the original exhaust valve as an inlet valve in order to keep the path or the heat and thus pressure loss from the superheater or Hansen heat exchanger to the working cylinder low.

- Since the original cams for the steam cylinder on the camshaft no longer fit the application, they must be replaced by suitable cams. However, the cams are very small and do not provide much valve lift. One conceivable way to remedy this is to give the steam cylinder its own camshaft, which rotates in the same axis on the existing camshaft, but has twice the speed, i.e. crankshaft speed, so that the cams can grip over twice the angular range and thus produce considerably more valve lift. The intake cam would then have the following shape, for example:



- If this is not necessary due to the design of the valve train, the following applies: The steam cylinder works in two-stroke mode, i.e. each revolution has a power stroke, unlike the four-stroke diesel cylinders. Because the camshaft is tuned to the four-stroke cycle, it turns only half as fast as the crankshaft. This means that for the steam cylinder, the camshaft receives two opposite cams per valve.

- For a new design, the crankshaft journals would also have to be readjusted in their position to each other. The reason is the now new mixture of two-stroke and four-stroke principles. An original three-cylinder engine corresponds to a four-cylinder engine, since there are a total of four power strokes for every two crankshaft revolutions or one camshaft revolution.

- The piston ring should also be replaced by a gapless one made of plastic, as this seals hermetically or better. This is possible because high combustion temperatures do not occur.

- The former combustion chamber must be filled as far as possible, as it would reduce the performance. It would be dead volume, so to speak.

- The valve spring and the inlet valve seal must be reinforced, as the valve must be able to withstand the differential pressure. $\Delta p = p_1 - p_2$ must withstand the differential pressure. Since pressures higher than p_1=26 bar are also to be expected, a constructively new valve is advisable in the case of a new design.

- To ensure lubrication of the components, a little silicone oil is added to the working medium.

7 Optimisation

The timing of the intake valve determines the performance of the steam cylinder. While the exhaust valve is open from BDC to TDC, the intake valve starts to open at TDC, but it is not obvious when it has to close.

7.1 Inflow work

Contrary to what one might expect, optimisation does not begin with performance $P = M \cdot \omega$ but with the work $W = F \cdot s$ [8]. because even if the mechanical work is rotational work, it is transformed from translational work. Frictional losses are neglected in this rough calculation. The force F is pressure p times the circular area A: $F = p \cdot A$. The distance s corresponds to the part of the stroke H that is used until the inlet valve closes: $s = x$. In order to obtain the work W in Nm, p is calculated in $\frac{N}{m^2}$, A in m^2 and s, x and H in m.

Inflow work $$W_1 = p_1 \cdot A \cdot x \qquad (2)$$

7.2 Expansion work

After the inlet valve closes in the working cycle, the working medium begins to expand. This expansion proceeds adiabatically, i.e. idealised without heat exchange with the environment, since heat exchange takes time and the process considered here proceeds quickly. However, since an expansion involves a cooling of the expansion medium, the pressure also changes disproportionately to the volume change in contrast to the isothermal process. The condition is $p \cdot V^{\kappa} = konst.$ [4]. the exponent κ for R12346ze(Z) has the value $\kappa = 1.142$ (cf. Chap. 4.1).

The volume V results from the circular area A of the piston head $V = A \cdot s$. In the case of the expansion work, the path s goes from the inflow depth x to the stroke H.

From $p \cdot V^{\kappa} = konst.$ becomes $p_2 = \frac{p_1 \cdot V_1^{\kappa}}{V_2^{\kappa}}$ respectively. $p_2 = \frac{p_1 \cdot x^{\kappa}}{H^{\kappa}}$, which is the pressure p_2 at the end of the expansion. However, the pressure in the cylinder changes constantly during the adiabatic expansion of the working medium, so that the work must be summed up in the form of an integral:

Expansion work $$W_2 = A \cdot p_1 \cdot \int_x^H \left(\frac{x}{y}\right)^{\kappa} dy \qquad (3)$$

Integral solved $$W_2 = -A \cdot p_1 \cdot H \cdot \frac{x^{\kappa}}{\left[\left(H^{\kappa}\right) \cdot (-1+\kappa)\right]} + A \cdot \frac{p_1}{(-1+\kappa)} \cdot x \qquad (4)$$

7.3 Outflow work

It can be assumed that the pressure in the low-pressure section corresponds to p2 in the worst case, since the steam condenser takes pressure away by cooling and liquefying the expansion medium or otherwise the capacity of the condenser is too low and the machine would not work - the steam volume would build up. After opening the outlet valve, the piston pushes the expansion medium against the pressure p2 so that the outflow work is similar to the inflow work:

Outflow work

$$W_3 = -A \cdot p_2 \cdot H$$
$$W_3 = -A \cdot p_1 \cdot \left(\frac{x}{H}\right)^{\kappa} \cdot H \qquad 5$$

7.4 Inlet depth xopt

With these relations, the total work Wges of a crankshaft revolution can be recorded:

Total work

$$\text{Wges=W2+W1+W3}$$
$$W_{ges} := -A \cdot p_1 \cdot H \cdot \frac{x^{\kappa}}{\left[\left(H^{\kappa}\right) \cdot (-1+\kappa)\right]} + A \cdot \frac{p_1}{(-1+\kappa)} \cdot x_i + p_1 \cdot x \cdot A - p_1 \cdot \frac{x^{\kappa}}{\left(H^{\kappa}\right)} \cdot H \cdot A \qquad 6$$

With x=0..H the following graph results:

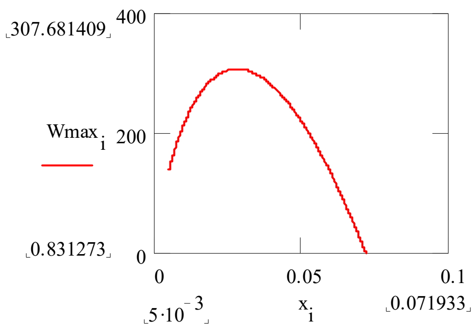


Figure 3: Course of overall work

The abscissa shows the inflow depth x, above which the total work Wges is listed on the ordinate. A maximum is evident. This can be found by setting the derivative to zero. However, the equation is too complicated to solve for x, and so it must be iterated using Newton's method:

In order to determine the convergence of the method, the following is determined $\left|\frac{f(x) \cdot f''(x)}{\left(f'(x)\right)^2}\right| < 1$ is determined. In this case, the initial x is 0.01 m. The check yields 0.961 and is thus <1, i.e. the process converges. Now the actual calculation can be carried out mechanically $\left(f(x) = \frac{W_{ges}}{A \cdot p_1}\right)$:

$$x_{opt} := \left| \begin{array}{l} \text{while } f'(x) \geq 0 \\ \quad \left| \begin{array}{l} a \leftarrow f'(x) \\ b \leftarrow f''(x) \\ x \leftarrow x - \frac{a}{b} \end{array} \right. \\ x \leftarrow x \end{array} \right.$$

The result for the optimal inflow depth xopt is $x_{opt} \approx 0.028\,m$. Since in all terms of W_{ges} [see Eq. 6] $A \cdot p_1$ occurs in all terms of Wges, the optimum inflow depth is only dependent on the stroke H. Thus it can be given as a percentage of the stroke (see chapter 5):

Optimum inflow depth [%] $$\frac{x_{opt}}{H} \cdot 100 = 39{,}25\,\% \qquad 7$$

7.5 Control times

In piston engines, the control time is usually not given as a function of the stroke, but in angles before and after TDC (example calculation: connecting rod length pl=110 mm 7:

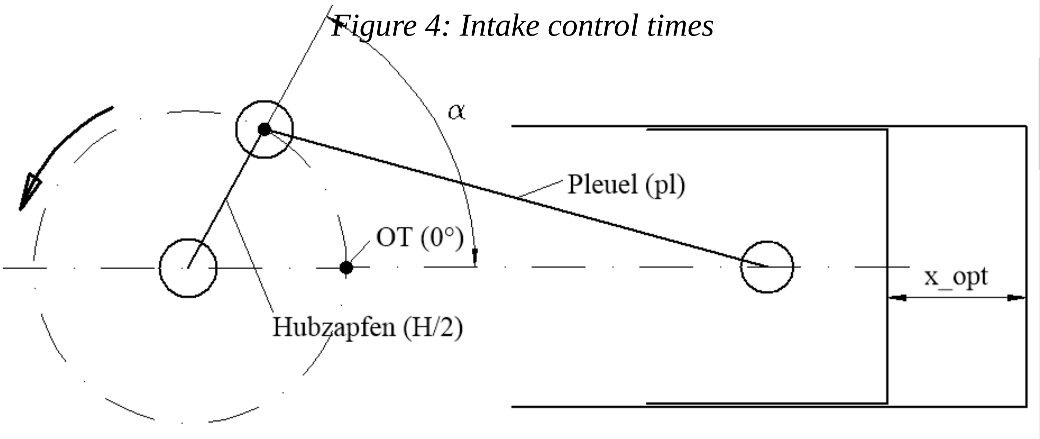


Figure 4: Intake control times

Inlet depth via crankshaft $$x_{opt} = \left(\frac{H}{2} + pl\right) - \left[\frac{H}{2}\cdot\cos(\alpha) + \sqrt{pl^2 - \left(\frac{H}{2}\cdot\sin(\alpha)\right)^2}\right] \qquad 8$$

Angle on crankshaft $$\alpha = \pi - \operatorname{acos}\left[\frac{\left[4\cdot\left(x_{opt}\right)^2 - 4\cdot x_{opt}\cdot H - 8\cdot pl\cdot x_{opt} + 2\cdot H^2 + 4\cdot H\cdot pl\right]}{\left(4\cdot x_{opt}\cdot H - 4\cdot H\cdot pl - 2\cdot H^2\right)}\right] \qquad 9$$

Auf der Kurbelwelle von OT bis $\alpha\cdot\frac{180}{\pi} = 68.854$ Grad nach OT

Auf der Nockenwelle von OT bis $\frac{\alpha}{2}\cdot\frac{180}{\pi} = 34.427$ Grad nach OT

8 Power

The maximum possible work per crankshaft revolution W_{ges} is, by substituting x_{opt} into Eq. 6 as can already be seen in Fig. 3 $W_{opt} \approx 308\, N \cdot m$. With this knowledge, a statement about the power can be made, since the speed is known:

$P_{Dampf} = \frac{W_{ges} \cdot n}{60000}$. The unit equation for this: $\frac{N \cdot m}{min} \cdot \frac{min}{60\, s} \cdot \frac{KNm}{1000\, N \cdot m}$ and $\frac{KN \cdot m}{s} = KW$.

At a pressure p_1 of 24.7 bar and a speed of 3600 min^{-1} , this results in a power P_{Dampf}=18.5 KW for the steam cylinder alone. However, it cannot be assumed that the pressure p_1 is the same at all speeds. Here we can only speculate for the time being.

However, it is already possible to make a statement about the maximum output, since the inventor Prof. h.c. Uwe Hansen determined the temperature of the working medium to be around 130 °C on his test machine, which was taken into account:

Only two cylinders remain from the diesel engine, so that their power results in $P_{Diesel} = 2/3 \cdot P_{Max} = 2/3 \cdot 20{,}3\, KW = 13{,}5\, KW$ results.

This results in a new maximum output of $P_{ges} = P_{Diesel} + P_{Dampf} = 32{,}0\, KW$.

9 Efficiency

The power of the steam cylinder decreases proportionally with the speed. It draws its power from the waste heat of the diesel cylinders. Thus, the total power cannot be above the calorific value H_u of the diesel fuel.

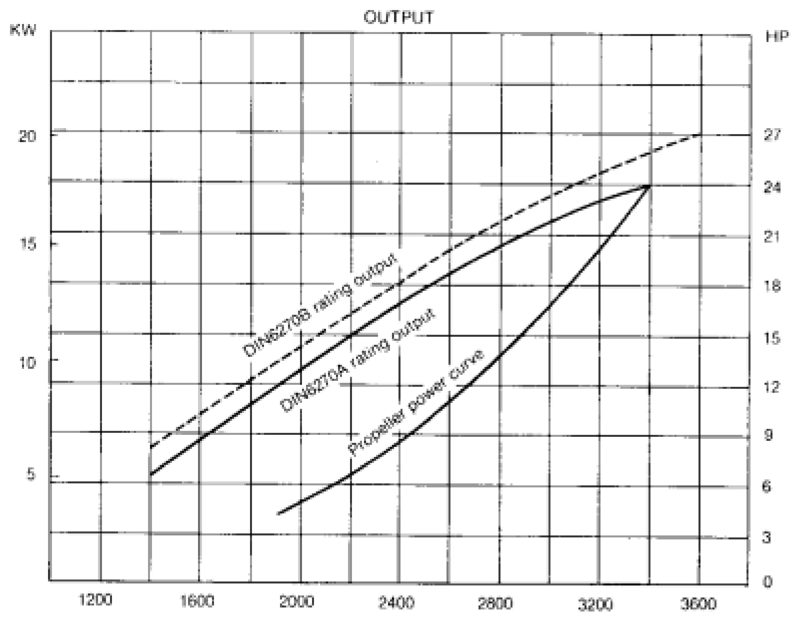
9.1 Consumption

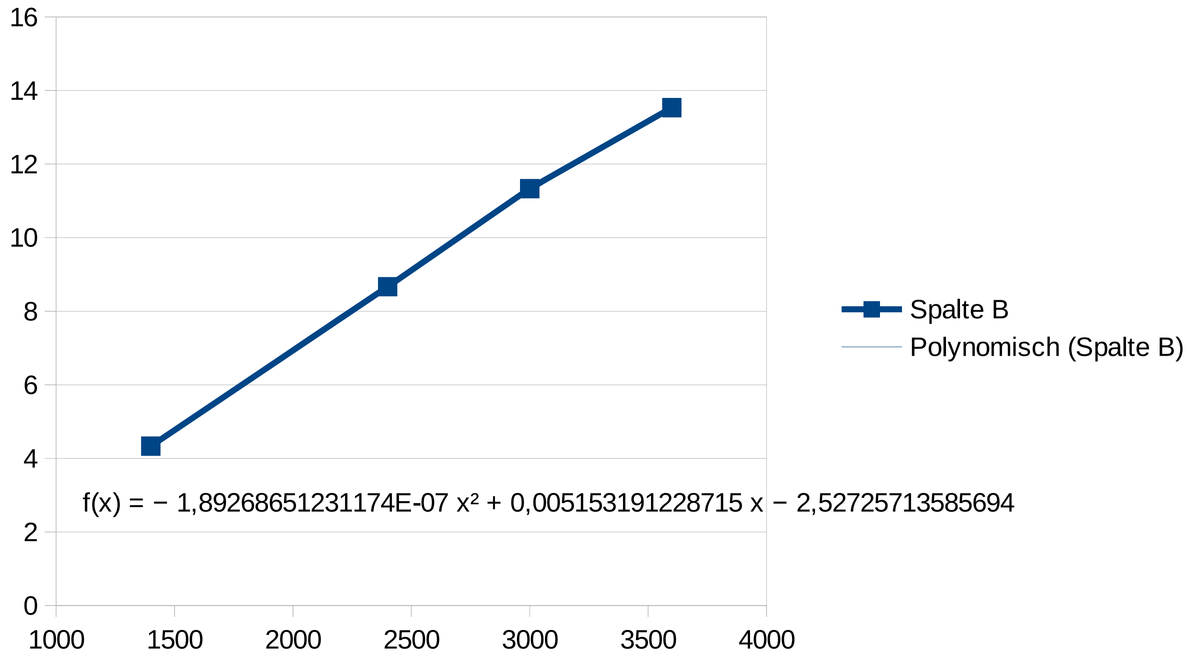
In order to make a conclusive statement about the consumption, it would be necessary to quantify the steam generation via demanded power and speed. Nevertheless, an indication should be attempted within the framework of the data available here.

Since the engine now only has two instead of three cylinders, which work according to the diesel principle, the fuel consumption cannot rise above 2/3 of the maximum consumption. Chapter 5 statements on this.

9.1.1 Power

First, a function is to be formed according to the diagram (see below) [7].This is then done using the spreadsheet programme:





1. Schaubild: Function P(n) Ex. engine, 2-cyl.

The function is intended to represent the power curve according to DIN 6270 B in two-cylinder mode, since consumption is also according to this standard. In principle, the power curve is a straight line. However, in order to make the calculation principle transparent, the complete function is used.

$P_{ges} = P_{Diesel}(n) + W_{ges} \cdot \frac{n}{60 \cdot 1000}$ (P_{Diesel}(n) corresponds to f(x) in diagram 1).

Before the conversion, the maximum power of the example machine was 20.3 KW at 3600 min^{-1}. For this power, at a pressure p_1 = 24.7 bar, after the conversion only

$$N(bar) := \frac{-1}{(2\cdot a2)}\cdot\left[a1 + \frac{Wges(bar)}{k} - \frac{1}{k}\cdot\sqrt{a1^2\cdot k^2 + 2\cdot a1\cdot k\cdot Wges(bar) + (Wges(bar))^2 + 4\cdot a2\cdot k^2\cdot Pt - 4\cdot a2\cdot k^2\cdot a0}\right]$$

n_{Vgl}=2318 [min^{-1}] is required.

9.1.2 Consumption

Following the example of chapter 9.1.1 the function for consumption is generated.

$$f(x) = 2{,}08011760382213E\text{-}05\ x^2 - 0{,}11635611907387\ x + 437{,}18118338846$$

Fixed at a total power P_{ges}=20.3 KW, the necessary speed v_p results for varying pressures (Pt = 20.3 KW maximum power of the original machine, k = 60000).

$$v_p := \frac{-1}{(2\cdot a2)}\cdot\left[a1 + \frac{Wges_p}{k} - \frac{1}{k}\cdot\sqrt{a1^2\cdot k^2 + 2\cdot a1\cdot k\cdot Wges_p + \left(Wges_p\right)^2 + 4\cdot a2\cdot k^2\cdot Pt - 4\cdot a2\cdot k^2\cdot a0}\right]$$

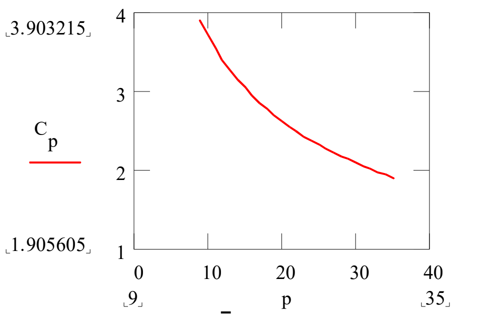
Thus, via the aforementioned consumption polynomial for the diesel share

$$C_p := \left[c2\cdot\left(v_p\right)^2 + c1\cdot v_p + c0\right]\cdot\frac{PV_p}{1000}$$

with the power of the diesel share ($P_{Diesel}(n)$ corresponds to PV_p)

$$PV_p := a2\cdot\left(v_p\right)^2 + a1\cdot v_p + a0$$

the consumption C_p at any pressure p_1 in kg/h above the pressure of the working medium is shown on the abscissa:



With the original machine, a power reduction of 20.3 KW corresponds to a consumption of 5.86 kg/h.

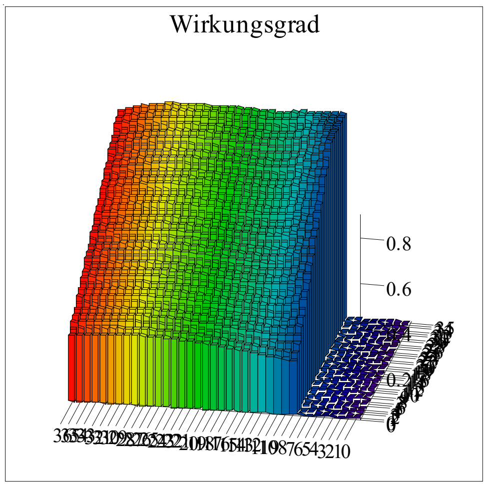
Although the pressure of the working medium was selected between 9 ... 35 bar for the Hansen engine, the reduction in consumption reaches at least 33 %. At 30 bar, fuel savings of more than 60 % are achieved.

9.2 Efficiency

The original machine has an efficiency of $\eta_{Original} = \frac{20{,}3\,KWh}{5{,}86\,kg \cdot 11{,}9\,\frac{KWh}{kg}} \approx 0{,}29$.

The efficiency of the Hansen engine between 9 ... 35 bar is dependent on the pressure of the working medium p and the speed n at a calorific value of the diesel fuel of $H_u = 11{,}9\,\frac{KWh}{kg}$ [9]:

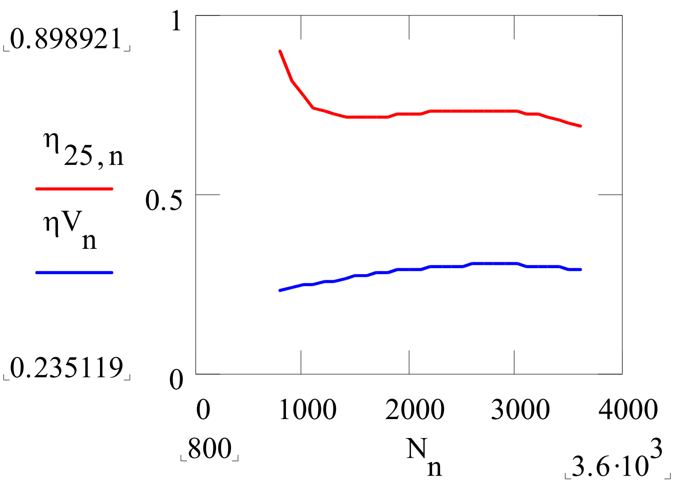
$$\eta_{p,n} = \frac{P_{ges,p,n}}{H_u \cdot C_n}$$



Eta

η is limited to 0.9.

This representation can be broken down in many ways. One example is the representation of ηn at fixed pressure:



The blue graph is the efficiency of the original machine. Above it in red is the Hansen motor at p_1 = 25 bar.

10 Final remark

With a power reduction of 20.3 KW and a pressure of the working medium of only 9 bar, this results in an efficiency of 45 %, which corresponds to an increase of 55 % compared to the original engine. According to the Clausius-Clapeyron equation, the temperature of the working medium required to achieve this efficiency is just 82 °C, so that this efficiency does not even require the operating temperature of the cooling water.

It can be assumed that the use of the Hansen engine will produce far higher efficiencies than modern diesel units will ever achieve. The diagrams show the best prospects for further research on the Hansen engine. In view of the high resource consumption of alternatives to the internal combustion engine that are being planned and implemented, the Hansen engine is enjoying a renaissance.

The operating temperature of the working medium of 130 °C stated by the inventor has yet to be proven in continuous operation. For a new design, however, an adapted diameter would be advisable for the steam cylinder. The ratio of steam cylinders to diesel cylinders has not been determined.

The discrepancy between the efficiency determined mechanically here and the thermal efficiency of the steam part $\eta_{th}=1-\frac{T_2}{T_1}=0.12$ is huge, but the thermodynamic calculation of harmonically designed machine components such as the evaporator, superheater and condenser is still pending, since steam generation is the decisive criterion here. Nevertheless, a determination of the overall efficiency by means of measurement technology on the practically created prototype already seems inevitable.

11 Appendix

11.1 Bibliography

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11.2 Formula symbol

A	Circular area πr^2
A	Coefficient Antoine equation
B	Coefficient Antoine equation
b	Coefficient Straight line equation
C	Coefficient Antoine equation
F	Force on piston crown
H	Hub
H_u	Calorific value diesel
m	Slope Straight line equation
OT	Top dead centre
P	Power
p	Print
p_1	Pressure before admission
p_2	Pressure after expansion
p_c	Critical pressure (maximum pressure)
P_{Diesel}	Diesel cylinder power
P_{ges}	Total output steam+diesel
pl	Connecting rod length
ps	Pressure in Antoine equation
p_{sp}	Pressure at boiling point (1.01325 bar)
p_{Tri}	Pressure at the triple point
s	Path
T	Temperature in Kelvin
T_1	Element of the C.C. straight line equation
T_2	Element of the C.C. straight line equation
T_c	Critical temperature
T_{sp}	Temperature at boiling point
T_{Tri}	Temperature at the triple point
UT	Bottom dead centre
V	Volume
V_1	Volume at inlet closure
V_2	Volume at UT
W	Work
W_1	Inflow work
W_2	Expansion work
W_3	Outflow work
W_{ges}	Total work
W_{opt}	Total work at xopt
x	Travel with open inlet valve
x_{opt}	Optimum path with open inlet valve
α	Control time on crankshaft
Δp	Differential pressure
η	Efficiency
$\eta_{Original}$	Diesel efficiency
ϑ	Temperature in °C
κ	Adiabatic exponent