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CONSIDERATIONS ON THE EXTERNAL COMBUSTION SYSTEM OF THE STIRLING HOT GAS ENGINE

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16. Abstract

After an introduction on the Stirling engine the external combustion system as well as the general loss division and efficiencies are described. The requirements for the combustion system and different variants of the combustion system are compared and discussed.

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CONSIDERATIONS ON THE EXTERNAL COMBUSTION SYSTEM OF THE STIRLING HOT GAS ENGINE

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1. Stirling Engine /1*

At the beginning of the previous century Robert Stirling (1816) invented a closed hot air process and its technical implementation with already all the decisive features of the present Stirling engine. The lacking material qualities and the want of knowledge of thermodynamics prevented the suitable layout and the breakthrough for this machine as compared with steam machines and combustion engines. In the thirties development studies on the Stirling engine started at the Company Philips in Holland (1). In Germany the Company MAN of Augsburg and MWM of Mannheim have started jointly working on the development of Stirling engines.

The Stirling engine is a piston machine with external heat supply and enclosed internal gas circuit process. The internal circulation medium is helium. The internal circulation process is controlled in such a way that a uniform heat supply can be achieved from outside, as though the engine were fed constantly from a heat tank with constant high temperature. The thermal flows fed to and removed from the internal process flow through heat exchangers (recuperators).

In the internal circular process of the Stirling engine in the ideal case a double isotherm-isochoric process takes place with helium (compare later Figure 2). On the isotherms of high temperature the heat is supplied, at low temperature the residual heat is removed from the process. On the isochores the heat is exchanged with high efficiency through regenerators. The actual process affected by losses takes place within the ideal one with high efficiency. Greater details on the internal circular process and technical implementation have been given in this journal by Maijer (1) on the Philips-Stirling engine.

*Numbers in the margin indicate pagination in the foreign text.

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The internal circular process is independent of the type of the external supply of heat because of the separating walls of the recuperators in the material flows. Bascially the Stirling engine is in the ordinary sense able to use thus many fuels because it only requires a heat generator with a certain temperature. All liquid fuel can be burned with air in an optimal manner in the same heating plant of the Stirling engine.

But in the external generation of heat a uniform continuous combustion is possible. Only the mobile parts of the power unit, auxiliary unit and the continuous gas flows contribute to the noise of the engine. Therefore the total noise level is much lower than for internal combustion engines. The uniform flame also allows a "clean" combustion.

2. The External Combustion System

The external production of heat for the Stirling engine is in the generally most convenient case achieved by external combustion. This is an open through process with air and combustion gases connected before the Stirling engine, Figure 1.

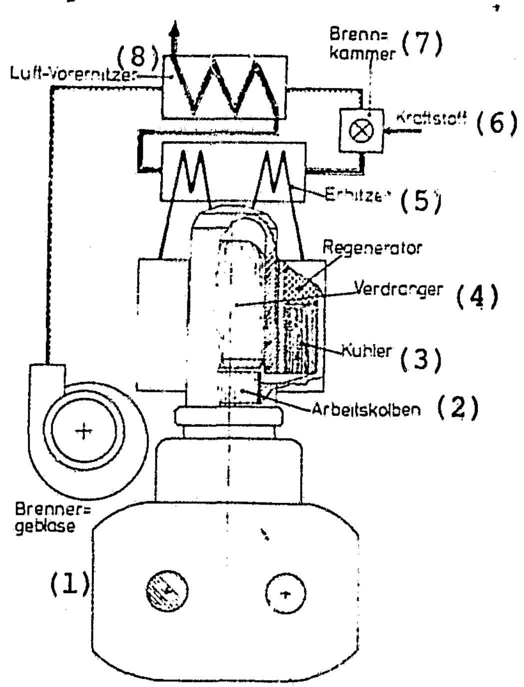

Figure 1: The Stirling engine with external combustion. Key: (1) burner blower; (2) working chamber; (3) illegible; (4) compressor; (5) heater; (6) fuel; (7) combustion chamber; (8) air preheater.

A blower conveys the necessary combustion air to the combustion chamber. After the combustion in the combustion chamber the hot combustion gases first activate the heater, that is the hot heat exchanger of the Stirling engine and give off the thermal energy there to operate the engine. Since the Stirling engine can no longer use directly the thermal content of combustion gases below about 700 degrees C, from 700 degrees C onwards the energy content of the combustion gases would be lost if an air preheater (preliminary heating device for the air, Luvo) were not able to recover the heat for combustion. /2

Figure 2 provides information on the thermal flows in the external and internal system of the Stirling engine, in which Mollier h,s diagrams of the internal and external process are plotted which are coordinated through the absolute temperature scale.

In the external heat generation continuous process the blower work to maintain the heating process, the effect of the air preheater, the combustion heat QB to be supplied and the heat QE to be removed in the Stirling process is recognizable. This heat must be transmitted as energy or heat flow Φ E through the heater to the internal cycle with helium, which is indicated in Figure 2 by the dashpoint reference lines.

Figure 2 shows on scale the changes in state. The entropy scale is interrupted; for entropy values both in the internal circular process with helium and in the external continuous process with air and combustion gases the entropy zero points $s = 0$ were chosen at 273.16 K at $p = 1$ bar. For helium the enthalpy-entropy diagram according to Bammert (2) and for combustion gases the diagram of Pflaum (3) were taken as bases.

To satisfy the heat QE required by the internal engine circulation with the thermal content of the combustion gases, as is apparent from Figure 2, very high gas and combustion chamber temperatures are needed.

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3. Loss Division and Efficiencies

On the way from fuel energy to effective mechanical power in energy conversion in the Stirling engine losses occur which can be described by a set of products of partial efficiencies just like for engines with internal combustion.

The total efficiency is the quotient of the effective shaft power and the fuel energy used:

$$\eta_o = P_e / \phi_B \qquad (1)$$

This overall relation can be subdivided as follows:

$$\eta_o = \frac{P_e}{P_{SAE}} \cdot \frac{P_{SAE}}{P_i} \cdot \frac{P_i}{P_v} \cdot \frac{P_v}{\phi_E} \cdot \frac{\phi_E}{\phi_B} \qquad (1a)$$

with

$$\eta_e = \eta_{mH} \cdot \eta_{mS} \cdot \eta_g \cdot \eta_v \cdot \eta_B \qquad (1b)$$

$\eta_B = \frac{\Phi_E}{\Phi_B}$ burner efficiency (heater thermal flow to the fuel energy used)

$\eta_v = \frac{P_v}{\Phi_E}$ efficiency of the total Stirling process (power of the perfect Stirling engine to the heat flow supply; Carnot efficiency)

$\eta_g = \frac{P_i}{P_v}$ quality of the internal Stirling process. Approximation of the real to the theoretical process (indexed piston power to the power of the perfect Sterling process)

$\eta_{ms} = \frac{P_{SAE}}{P_i}$ mechanical efficiency of the Stirling engine without auxiliary equipment (mechanical power on the crankshaft without mechanical losses show auxiliary equipment for the index piston power)

$\eta_{mH} = \frac{P_e}{P_{SAE}}$ mechanical efficiency because of lost powers through the auxiliary equipment (effective drive power to crankshaft power without losses to auxiliary equipment).

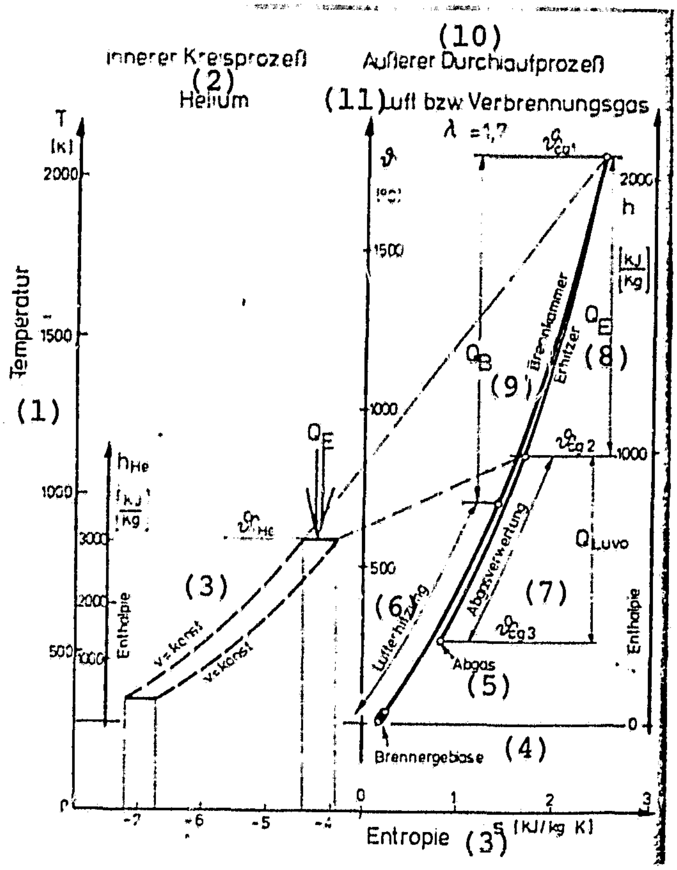

Figure 2: Mollier h,s diagrams of the internal and external process. Key: (1) temperature; (2) internal circular process; (3) entropy; (4) burner blower; (5) waste gas; (6) air heating; (7) waste gas heating: (? illegible); (8) heater; (9) combustion chamber; (10) external continuous process; (11) air or combustion gas.

To evaluate the external combustion plant an efficiency must be taken which consists of the burner efficiency and the burner blower component of the mechanical efficiency as a result of loss powers through auxiliary equipment, since the burner blower is the component of the combustion system.

If PG is the power of the burner blower, the efficiency of the external heating system is defined as follows:

$$\eta_{ER} = \eta_B \cdot \left(\frac{P_{SAE} - P_G}{P_{SAE}}\right) = \frac{\phi_E - (P_G/\eta_S)}{\phi_B}$$

with $\eta_S = \eta_V \cdot \eta_g \cdot \eta_{mS}$, the degree of conversion of the internal Stirling engine. Through the splitting of the blower power PG from the efficiency

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mE this efficiency improves to:

$$\eta_{mH \cdot G} = \eta_{mH} \left(\frac{P_{AE}}{P_{AE} - P_G} \right) \qquad (3)$$

The total efficiency is then unchanged:

$$\eta_e = \eta_{mH \cdot G} \cdot \eta_{mS} \cdot \eta_g \quad \eta_v \quad \eta_{ER} \qquad (4)$$

Since the efficiency of heating η_{ER} (? illegible) is included directly in the total efficiency, it is equally important as the other partial efficiency.

4. Requirements for the Combustion System /3

4.1 Starting

The external combustion plant is an automatic system and must be started itself before the starting of the internal circulation and the Stirling engine, in order to preheat all the heat exchanges and to have sufficient heat available for the starting of the Stirling engine. For this reason the burner blower and a fuel delivery pump are first operated electrically from battery. Thereupon the Stirling engine can be started. When the Stirling engine is brought to idling speed, the auxiliary equipment indicated are taken over and driven mechanically by the engine through the override clutches. The total starting processes last according to experience less than a minute.

4.2 Operation for Constant Load

The temperatures θ_{cq1} and θ_{cq2} as well as the amounts of combustion gas of the external system must be adjusted to the energy requirement of the engine Φ_E and specifically in such a way that according to the laws of thermal transfer the required heat flow Φ_E is transmitted to the hot heat exchanger to the internal cycle. The heat transfer relation is:

$$\Phi_E = k \cdot A_{Ea} \cdot \Delta t_m \qquad (5)$$

in which

$$k \cdot A_{Ea} = \frac{A_{Ea}}{\left(\frac{A_{Ei}}{\alpha_i} + \frac{\delta \cdot A_{Em}}{\lambda_W} + \frac{1}{\alpha_a} \right)} \qquad (5a)$$

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and

$$\Delta t_m = \frac{\vartheta_{cg1} - \vartheta_{cg2}}{\ln\left(\frac{\vartheta_{cg1} - \vartheta_{iHe}}{\vartheta_{cg2} - \vartheta_{iHe}}\right)} \qquad (5b)$$

with the designations of Figure 2.

The internal heat transfer coefficient in the heater

$$\alpha = C_i (Re_{Ei})^{0.8} \qquad (5c)$$

and the external heat transfer coefficient in the heater

$$\alpha_a = C_a \cdot (Re_{Ea})^{0.5} \qquad (5d)$$

are calculated through the Reynolds number.

The internal flow conditions in the heater are pregiven by the Stirling engine. From this follow the values for Re_{Ei} and the maximum temperature θ_{iHe} of helium. The heat transfer coefficient a is therefore adjusted according to the operating conditions of the engine, the load and speed.

For the external combustion gas flow we have

$$Re_{Ea} = \frac{\dot{m}_{cg} \cdot d_E}{\bar{\eta}_{cg} \cdot S_{Ea}} \qquad (5e)$$

in which S_{Ea} is the external passage cross-section for the combustion gas through the heater and dE the diameter of the tube of the heater.

With these quantities we have

$$\phi_E = \frac{A_{Ea}\left(\frac{\vartheta_{cg1} - \vartheta_{cg2}}{\ln((\vartheta_{cg1} - \vartheta_{iHe})/(\vartheta_{cg2} - \vartheta_{iHe}))}\right)}{\left(\frac{A_{Ei}}{\alpha_i} + \frac{\delta \cdot A_{Em}}{\lambda_w} + \frac{1}{C_a\left(\frac{\dot{m}_{cg}}{\bar{\eta}_{cg}} \cdot \frac{d_E}{S_{Ea}}\right)^{0.5}}\right)} \qquad (6)$$

Simultaneously the cooling of the combustion gas is:

$$\phi_E = \dot{m}_{cg}(h(T_{cg1}) - h(T_{cg2})) = \dot{m}_{cg} \; c_{pmcg} \, (\vartheta_{cg1} - \vartheta_{cg2}) \qquad (7)$$

with cpmcg as average of the specific thermal capacity of the combustion gas, valid for the temperature range θ_{cg1} to θ_{cg2}.

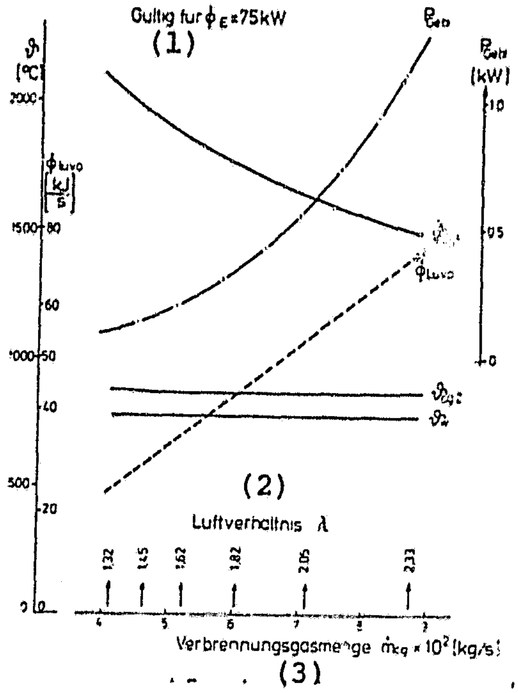

Figure 3: Temperatures and energies in the combustion plant as a function of the rate of flow of gas. Key: (1) valid for; (2) air ratio; (3) amount of combustion gas.

In the solution according to the temperatures we obtain the implicit equation (8) which can be solved by iteration with the valid material quantities for example from (5):

$$\vartheta_{cgl} = \vartheta_{iHe} + \frac{\phi_E}{\dot m_{cg} \cdot c_{pmcg} \cdot (1 - e^{-y})} \qquad (8)$$

with

$$y = \frac{(A_{Eo}/(\dot m_{cg} \cdot c_{pmcg}))}{\left(\frac{A_{EL}}{\alpha_i} + \frac{\delta \cdot A_{Em}}{\lambda_w} + \frac{1}{C_a} \cdot \left(\frac{\eta_{cg} \cdot S_{Eo}}{\dot m_{cg} \cdot d_E}\right)^{0.5}\right)} \qquad (8a)$$

Here there is a clear assignment between the maximum temperature of the internal Stirling process θ_{He} and the maximum combustion temperature θ_{cgl}.

A certain value of the heat flow Φ_E may be achieved with different amounts of combustion gases, while with the increasing amount of combustion gas the temperature θ_{cgl} needed for the heat transfer decreases, Figure 3.

Simultaneously however the necessary blower power PGebl for the maximum rate of flow and the consumption in the air preheater increases to be able to transfer the increasing thermal flow Luvo = QLuvo. mcg for the same loss of waste gas, that is the transfer area of the air preheater must be increased.

From a consideration of the exergetic efficiency of the combustion chambers of Baehr (4) it is also clear that the smallest possible amount of combustion gas mcg must be chosen for a satisfactory efficiency of the combustion plant.

Small amounts of combustion gases allow the limitation of the construction cost in the air preheater and a low blower power. Here however the maximum combustion temperatures $\bar{\theta}_{cg}$ (? illegible) are adjusted. High combustion gas temperatures and low amounts of combustion gases mean a low air proportion λ which should lie in the neighborhood of the stoichiometric combustion. The combustion chamber requires however a certain excess of air to achieve a complete combustion. A sure operation of the combustion /4
chamber without unburned components in the waste gas is according to experience assured with air proportions of $\lambda = 1.3$ to 1.5. This range of air ratio should tend in the sense of high heating efficiencies.

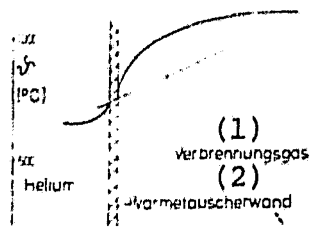

Figure 4: Heat transfer conditions on the heater. Key: (1) combustion gas; (2) heat exchanger wall.

The side of the heat exchanger walls of the heater turned towards the hot combustion gases has a very high temperature, which however is almost independent of the chosen air proportion and the amount of combustion gases, compare θ_W in Figure 3.

In Figure 4 the heat transfer proportions of the heater of the Stirling engine are shown. Because of the very satisfactory heat transfer for helium on the internal side of the heater the external wall temperature can be maintained also within technologically achievable limits.

4.3 The Combustion System with Changed Load

The internal circulation process of the Stirling engine with helium is regulated through the average cycle pressure. If the engine is to give off more power, then the pressure level must be increased by supplying helium. The heat flow Φ_E to be supplied must undergo a suitable increase.

An attempt will be made to maintain constant the maximum temperature of the internal circulation process even with unchanged load with regard to the efficiency of the process on one hand and the maximum permissible wall temperatures of the heater on the other in the entire load range. The external combustion system consequently must be readjusted as exactly as possible in load variations in the power output $\dot{\Phi}_F$ to satisfy the needs of the engine.

If in the regulating process slight fluctuations are allowed for the maximum temperature of the helium circulation process, then the thermal capacity of the heater walls act as an intermediate buffer, so that a sure phase displacement is allowed for the reregulation of the combustion system.

To obtain optimum conditions in the external continuous process in the entire load range, we must always proceed with the minimum possible amount of combustion gas, high combustion temperatures and low air proportion, compare Figure 3.

To this end it is necessary that the amount of fuel and air should be regulated simultaneously to a constant low air proportion which is achieved by throttling the fuel flow and the suction force of the burner blower.

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5. Different Variants of the Combustion System and Comparison

As a result of the extensive freedom in the process management in the external combustion system different variants are possible of which three types will be discussed here with the effects.

The circuit pictures are compared with the corresponding changes of state in Mollier h,s diagrams in Figure 5:

I Mechanically driven burner blowers and single stage combustion.
II Mechanically driven burner blowers and two or multi-stage combustion.
III Burner blower designed as turbochargers with single stage combustion (saving of blower power PGebl).

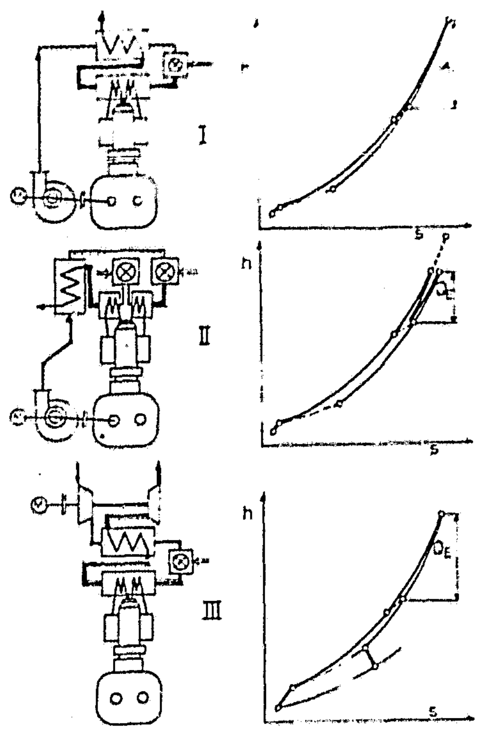

Figure 5: Types of variants of combustion plant.

For the previously built test engines the variant I was implemented.

The variant type II promises a better adjustment of the heat generation and the heat removal from the Stirling engine. The maximum combustion temperature ϑ_{cg} (? illegible) can be much lower. But several combustion chambers and a very complicated regulating system are required. The efficiency η_{FR} can be brought to comparable values of the variant I and differs only through the somewhat higher blower power, because of the higher pressure losses, in several combustion chambers. A special thermodynamic calculation was omitted because there are too few striking differences from variant I.

For the turbocharge variant III the operating conditions are calculated thermodynamically and compared with similar calculations for variant I.

To this end the external continuous process was constituted and simulated on an electronic data processing plant using the heat transfer laws for the heater and the air preheater. For the processing with the electron data processing unit the thermodynamic properties of the combustion gases and a program system according to (5), the transport quantities (viscosity, conductivity) according to Brandt (6) were used.

In the variant III it must be established whether:

1. the operation is possible with a waste gas turbocharger,
2. thermodynamic and structural advantages may be expected and,
3. how the turbocharger group is introduced in the regulating requirements of the external combustion system.

The turbocharger has been adjusted for the calculations in such a way that it applies in the full load point of the Stirling engine a pressure ratio of 3 to 1 in a single stage. The group efficiency for the unit quantities considered were introduced with a flow volume of about 0.055 m3/s at 0 degrees C and 760 torr with $\eta_{ATL} = (\Delta h_v/\Delta h_T)_s \approx 0.5$ according to (7). /5

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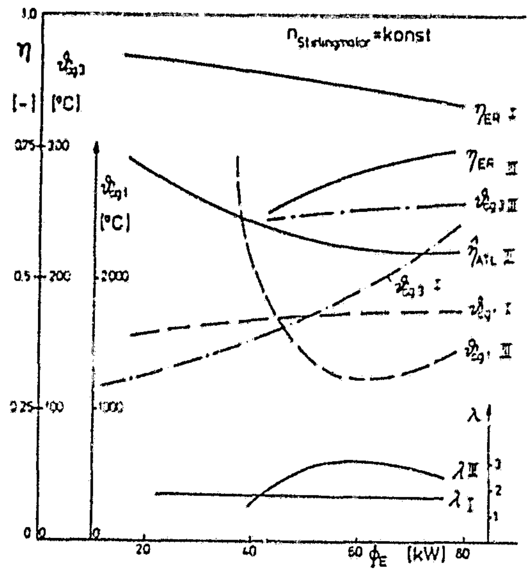

Figure 6: Comparison of efficiency and temperature of the combustion ones I and III.

The first calculations show that the combustion plant with a turbo- charger can only be operated if the air preheater is reduced to such an extent that the working capacity of the combustion gas is sufficient to cover the compression power including the turbocharger losses.

The comparison of the variants I and III from the point of view of layout gives for Φ_E = 75 kW:

Weight gas temperature:

$\vartheta_{cg\,3\,I} = 240°\,C$
$\vartheta_{cg\,3\,III} = 420°\,C$

Efficiency:

$\eta_{ER\,I} = 0{,}84$
$\eta_{ER\,III} = 0{,}73$

Accordingly the use of the turbocharger gives no thermodynamic advantage even from the point of view of design.

To control the behavior of a waste gas turbocharger group for partial load a plant of the first type was compared with a plant of type III for same powers Φ_E, Figure 6.

The best efficiencies of the combustion system are obtained in the entire load range of the engine (engine speed = constant) for mechanically driven burner blower η_{ER1}. Here the efficiency even increases further in the direction of partial load. The corresponding combustion temperatures θ_{cg11} and θ_{cg31} as well as the air proportion $\lambda 1$ show the favorable variation discussed in the previous section.

The turbocharger efficiency required for the same optical amounts of gases is shown in Figure 6 as η_{ATLIII}. Accordingly for partial load unachievably high turbocharger efficiencies are needed.

For comparison a plant of type III was taken which is equipped with an air preheating plant reduced to half and with a turbocharger of η_{ATL} = 0.6, to be able to advance as far as possible in the partial load range with the turbocharger arrangement. The efficiency here is moderate in the upper load range, because the charging group conveys such a large amount of air. But below the half-load the conveyor power is too low to apply the required thermal current Φ_{E}, Therefore the combustion temperature $\bar{\theta}_{cg1}$ rises quickly under half-load to temperatures which can no longer be controlled by the combustion chamber, whereas the air proportion λIII decreases strongly.

The quadratic delivery characteristic of the loading group can be adjusted only poorly to the linear regulating requirements of the combustion system in the Stirling engine. The use of the waste gas energy is better for an air preheating device than for an expansion in a turbine.

Moreover the use of the waste gas turbocharger does not represent any simplification in the construction cost of the combustion system.

The variants I and II tend to higher efficiencies, while a plant on the basis of II must maintain large power units because of the increased regulating costs.

A combustion plant according to I is preferred for assembly and traction Stirling engine sizes under thermodynamic and economic aspects.

Formula Symbols Used

Designations and units according to DIN 1345 (West German Standards).

C	Empirical constant	W/m2K
A	Heat exchanger surface	m2
P	Power, energy	W
Q	Heat	J
Re	Reynolds number	--
S	Flow cross-section	m2
T	Absolute temperature	K
cp	Isobar specific thermal capacity	J/kg K
d	Characteristic dimension, tube diameter	m
h	Specific enthalpy	J/kg
k	Thermal transfer coefficient	W/m2 K
m	Gas mass flow	kg/s
n	Speed of rotation	rpm
p	Absolute pressure	N/m2 or bars
s	Specific entropy	J/kg K
Φ	Thermal current, power, energy	J/s = W
α	Thermal transfer coefficient	W/m2 K
δ	Thermal exchanger wall thickness	m
$\bar{\eta}$	Bar viscosity of the gas	kg ms
η	Efficiency	--
ϑ	Celsius temperature	degrees C
λ_W	Thermal conductivity factor	W m K
λ	Air proportion	--

Indices

ATL	referred to waste gas turbocharger
B	referred to fuel

E referred to heater
ER referred to total heating system
G=Ge 1 referred to burner blower
SAE referred to power without losses through heating equipment
T referred to turbine
V referred to compressor
a external
cg referred to combustion gas
e effective
i internal
m average
s for constant entropy
v total

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