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A NEW INTEGRATED POWER PLANT WITH A SMALL SCALE TURBINE FOR THE ORGANIC RANKINE CYCLE

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Combined heat and power (CHP or cogeneration) describes the simultaneous generation of electrical power and heat. CHP has been well established for medium and high power ranges (> 100 kW el. power). The Kompakte Dampf Turbine (KDT, meaning compact steam turbine) addresses the low-end of power generation (~2 kW el. power). The KDT is a highly integrated power plant of small dimensions able to use various heat sources. Its simple design promises a low-cost CHP for residential homes.

Key words: integrated power plant, turbine, thermodynamic losses, generation.

The KDT is essentially a small, fully integrated power plant that offers a number of advantages as compared to other small scale systems: Very few parts, no valves and no feed pump. The simple build-up permits an economic construction and low maintenance costs. For lots produced on an industrial scale a price of Ђ1000 — Ђ2000 per unit is anticipated. This follows the tenet that new approaches for small scale systems have to be developed [2]. Current research focuses fluids following the Organic Rankine Cycle (ORC) [1; 2] because they offer some advantages over water. The aim is to reach at an overall electrical efficiency of 10% which would provide 1.5—2 kW of electrical power for a typical residential home. Once successful ORC prototypes have been developed the use of water will be investigated.

The Fundamentals of the small scale turbine (KDT)

The KDT was originally invented at the Hochschule Darmstadt [3]. The current prototype is shown in Figure 1. It is built up of a rotating cylinder (270 mm in diameter and 300 mm in height) and a static axis, therefore reversing the traditional arrangements in a turbine. The cylinder is divided by a plate into boiler and cooler and partially filled with an organic working fluid. The dividing plate is fixed to the cylinder therefore co-rotating with it. A small gap exists between the rim of the dividing plate and the cylinder, allowing the working fluid to pass through it. Due to the rotation and the resulting centrifugal forces the fluid rises up the inner wall of the cylinder and so seals the gap. Laval-nozzles are fixed to the dividing plate. The working cycle is similar to that of a traditional turbine. The heat transferred into the boiler results in a partial evaporation of the working fluid which is accelerated through the Laval-nozzles. Since the Laval nozzles are fixed to the dividing plate their thrust is transferred to the rotating cylinder providing the first stage in the power generation. Further power is generated by a suitable arrangement of blades similar to a traditional turbine, only the roles of blade and guide wheel are interchanged. The heat exchanger in the cooler leads to condensation of the expanded steam. The fluid is spun to the rotating cylinder wall and flows through the gap between the plate and cylinder back into the boiler, closing the thermodynamic cycle.

The rising of a fluid in a rotating cylinder is known as Newton's bucket. The complete formulas of fluids in rotating cylindrical cavities are derived by Gerber [4]. For very high rotational speeds the fluid effectively forms an annulus, the surface being plane for all practical purposes. The boiler and cooler are insulated from each other because the fluid covers the gap. This allows for different pressures in these chambers, a necessary prerequisite for a power plant. The concept of the KDT shown in Figure 1 is generalized allowing for boiler and condenser cylinders with different diameters as shown in Figure 2.

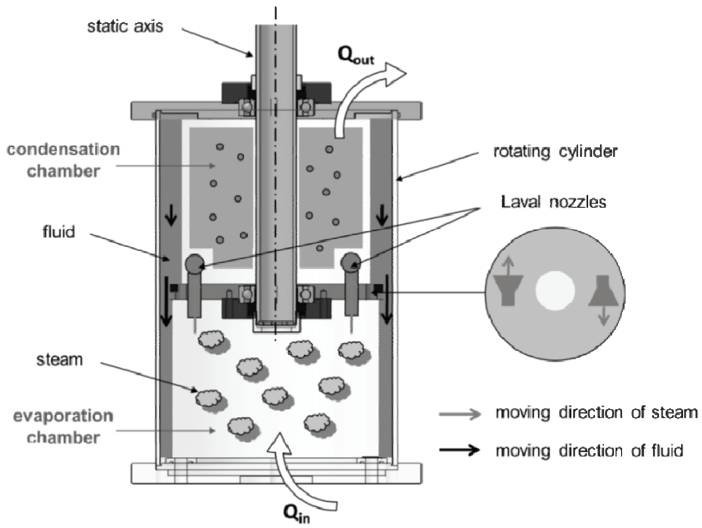


Figure 1. Schematic setup of the KDT in sectional view

The design in Figure 2 allows for higher pressure differences between boiler and cooler as the one shown in Figure 1. Using the results from Gerber [4] the gas pressures in the boiler and the condenser can be related to the various system parameters by:

$$p_2 - p_1 = \frac{1}{2}\rho_1\omega^2\left(r_4^2 - r_1^2\right) - \frac{1}{2}\rho_2\omega^2\left(r_4^2 - r_3^2\right). \tag{1}$$

Equation (1) describes the most general case with different radii for boiler and cooler. The requirement for two separate cavities is ensured by the 300 year principle of Newton's bucket. Another aspect of a power plant is its heat transfer ability. Here the KDT also relies on well the established research for Rotating Heat Pipes (RHP). Heat Pipes have been developed since the 1960s while RHPs have been researched since the 1980s. In Figure 3 the sketch of a RHP is given.

Upon description of a RHP it will become clear that it is very similar to the KDT as far as heat and mass transfer is concerned. A heat pipe has an evaporation and a condensing area. It is filled with some suitable working fluid and the heat is applied externally. The vaporized working fluid flows from the evaporator (high pressure) to the condenser (low pressure). Upon the condensation the working fluid is transported back to the evaporator. In traditional heat pipes this is done through the capillary forces in the wall materials. However, a RHP uses the centrifugal forces to transport the working fluid back to the evaporator. Here lies the strong similarity the KDT. For RHP the condenser side is equipped with a tapered angle that improves the mass flow characteristics. The major difference between a KDT and a RHP is the dividing plane which allows for much higher pressure differences in the KDT, especially in association with a design from Figure 2.

Design and Engineering considerations

The fundamental idea of the KDT is to create a tightly integrated system with low-maintenance requirements. The integration together with the small dimensions creates some design challenges. As the system is targeted for residential use the reference calculations are done with 20kW thermal input. Depending on the application there are different ways to transfer the heat into the KDT: 1) An external burner heating the outer surface of the KDT 2) a heat exchanger inside the boiler 3) Solar radiation through a window in the boiler. For the cooler a heat exchanger is necessary that is able to transfer approximately 90% of the energy into the cooling fluid. Measurements and theoretical studies for rotating heat exchangers show higher performance than static ones [5; 11].

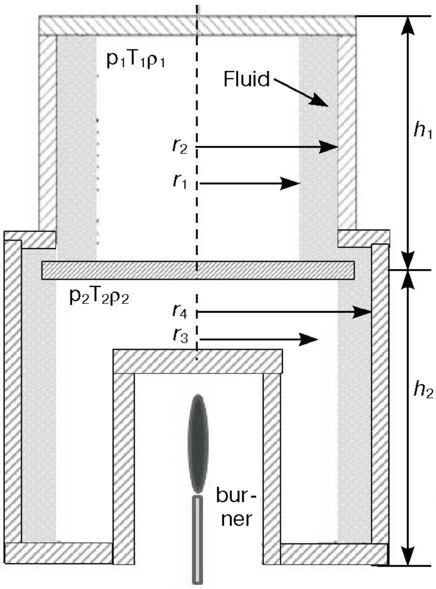


Figure 2. A generalized form of the KDT.
The lower part contains the boiler
the upper part the cooler

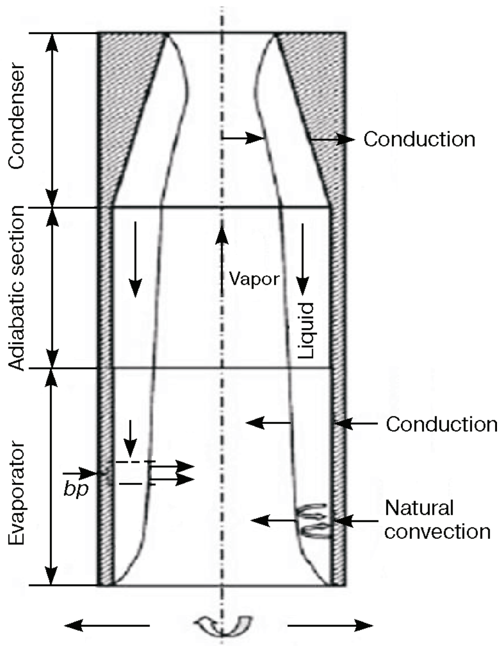
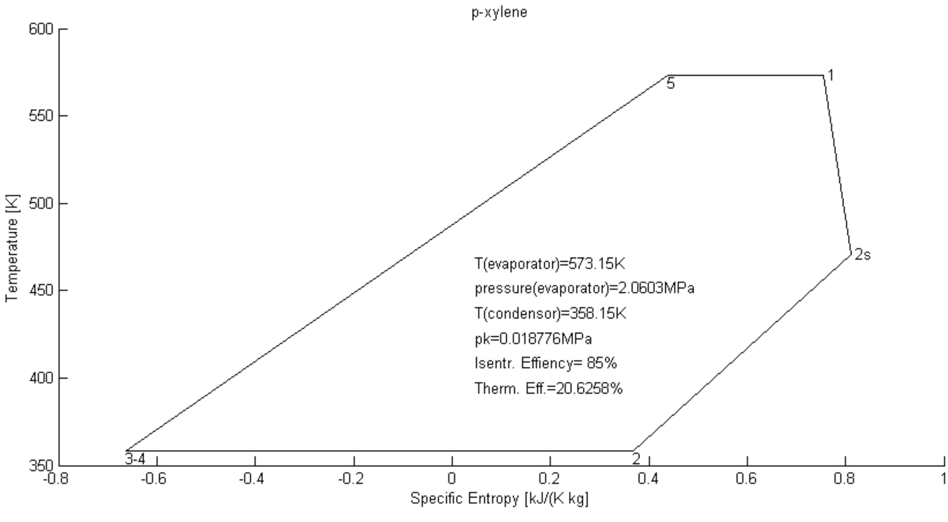


Figure 3. Sketch of a RHP from Song et al. [5])

The inlet and outlet tubes for the fluids of the secondary sides of the heat exchangers must be carried through the hollow static axis. This setup uses a special seal that creates a tight, sliding connection between the pipes in the axis and the rotating condenser. Initial calculations showed that the ideal rotational frequency is about 1000 rad/s Tests with the current prototype showed a bending eigenfrequency at about 300 rad/s, but it was possible to pass this point without any problems. With changing designs and filling rates further tests have to be made in order to find out if the existing system damping is sufficient or if additional measures have to be taken for safe operation. The material for the cylinder, bottom-, top- and dividing plate in the prototype is aluminum. This has to be changed for future prototypes as the strength requirements increase significantly. Ni-base alloys are considered as potentially useful due to their good mechanical properties such as strength and strain capacity over a wide range of temperature. Another very important aspect in the choice of material and the design is corrosion resistance. Especially at high temperatures this aspect has to be considered carefully. Uniform corrosion, intercrystalline corrosion, stress corrosion cracking or corrosion fatigue must be prevented as well as high temperature corrosion over the expected life time.

Thermodynamic aspects for the KDT with the example p-xylene

Recent years have seen a strong rise in the use of ORC fluids [6], especially in the area of low waste heat applications. They offer basically two advantages. Firstly they have high vapor pressures at relatively low temperatures and there are many "dry" fluids among them. This means that no superheating is necessary at the moment. The thermodynamic properties for p-Xylene are calculated using Fluidprop with Stanmix [7]. They can be compared readily with the work done by other researchers [8; 9]. The resulting TS diagram is shown in Figure 4.



State	Enthalpy kJ/kg]	Entropy [kJ/(K kg)]	T [K]	Pressure [Mpa]	Speed of sound [m/s]
1	406	0.7554	573.15	20.6	
L	393	0.7554	555.91	13.4	163
2s	261	0.8107	470.83	0.019	195
2	78	0.3694	358.15	0.019	170
3	−291	−0.6598	358.15	0.019	
4	−289	−0.6623	358.15	20.6	
5	224	0.4382	573.15	20.6	

Fig. 4. T-S diagram p-xylene. The calculations are normalized to 20 kW input

With these data it is possible to calculate the mass flow. The enthalpy difference results in a velocity of 539 m/s which corresponds to Mach number of 2.75. Due to the high velocity of a Ljungström wheel the Mach number at the entrance of the wheel is M = 1.46 for an angular velocity of 991 rad/s. Using a simple conical model for a Laval nozzle [10] the dimensions of the nozzle can be calculated. Since the size of the nozzle depends on the massflow it is in general advisable to work with multiple nozzles. For a configuration with 8 nozzles we find that they should have a length of 15 mm and an exit diameter of 4.1 mm.

In the future it is planned to investigate other fluids. One promising candidate is Butylbenzene [9; 8]. It offers the advantage that is has similar thermal efficiencies at lower vapor pressures which in turn benefit the dimensions of the KDT.

Cooler and boiler — estimations for the heat exchangers

So far the mechanical and thermo dynamical aspects of the KDT have been discussed. The most challenging part however is the design of the heat exchangers. On the boiler side app. 20kW have to be transferred into the KDT and on the cooler side 17 kW have to be transferred. out again. Let's consider first the cooler side of the engine.

With an admission of $\varepsilon = 0.25$ and a total surface of 0.12 m^2 the effective area is 0.03 m^2. The temperature difference between the fluid beam and the cooler is about 100 K. The important quantity is the heat transfer coefficient $\boldsymbol{\alpha}$. Measurements carried out by Hashimoto et al. [11] for a RHP found values for α of app. 8000 W/m^2K for methanol or water. The centrifugal forces are similar to those in the KDT. An alternative calculation according to VDI Wärmeatlas [12] for pipes results in a heat transfer coefficient of about 1340 W/(m K) for p-xylene at 465 K. The characteristic Length is $L = (\eta F^2/(\rho F^2 a))^{(1/3)}$ with the acceleration $\omega^2 r$, $Nu_{F,l} = 0.7\ Re_F^{-0,29}$ and $Nu_{F,t} = 0.0283\ Re^{-0.292} Pr_F^{0.333}/(1 + 9.66 Re_F^{-0.375} Pr_F^{-0.167})$ and $Pr_F = \eta_F\ c_{p,F}/\lambda_F$. Using these values of $\eta_F = 0.240$ mPas, $c_{p,F} = 2110$ J/kgK for liquid p-xylene the Nusselt number for the fluid is $Nu_F = (Nu_{F,l} + Nu_{F,t})^{0,5} = 0.0157$. Using $\alpha = Nu_F\ \lambda_F/L$ results in value of about 1340 $W/(m^2K)$. Due to the fact that the estimation are very crude we can only estimate the value of α for condensations somewhere in the range of about 1000 $W/(m^2K)$ to 8000 $W/(m^2K)$. Assuming the smaller value for $\alpha = 1340\ W/(m^2K)$ and ΔT of about 100 K and a stack of 6 thin fins with $A = 0.12\ m^2$ could result in a Q of about16kW. Other work on RHP [5] found values of 100—200 kW/m^2 in a RHP again leading to the right magnitude of heat transfer in the cooler.

In the boiler the heat exchanger must be able to transfer 20 kW into the KDT. Applying the model of Hottel [13] yields an energy transfer $Q = A\sigma\varepsilon_w/(1 - (1 - \varepsilon_w)(1 - A_v))\ (\varepsilon_g T_g^4 - A_v T_w^4)$. Applying some corrections this results in a transfer of about 14 kW inside the cylinder with the parameters: $T_w = 673$ K, $T_g = 1582$ K, $\varepsilon_w = 0.8$ and $A_v = 0.85$. Hereby we correct $\varepsilon_g = \varepsilon_g(p, T_g, s_g p_g)$ where P is the pressure and p_g is the partial pressure. And s_g is a correction factor which is calculated to $s_g = 0.9V/A$. $A_v = A_v(p, T_g, T_w, s_g p_g)$. The calculation steps are following the simple model of Hottel [13]. Only H_2O-Gas and CO_2-Gas is used for the estimation. Finally a small correction is applied and $\varepsilon_g = \varepsilon_{H20} + \varepsilon_{CO2} - (\Delta\varepsilon)_g$ and to $A_v = A_{v,\ H2O} + A_{v,\ CO2} - (\Delta\varepsilon)_w$.

The remaining 6 kW in the gas have temperature of about 680 K assuming a mean c_p of 1260 J/KgK and a dm/dt of about 0.0125 kg/s. This gas flux enters the small gap at the outside of the lower cylinder. Here we us the model for Tachibana for heat transfer of rotating cylinder to a gas. It is assumed that it is valid also for heat transfer from the gas to the cylinder. On the outside we apply the results of Tachibana [14] and these results in high α and a heat transfer of about 6 KW. For a inner rotating cylinder with irregular vortices we apply $\alpha\delta/\lambda = 0.046$ (Ta2Pr)1/3. This formula represents an empirical fit. Further for the heat transfer by the axial flow we apply

$$\alpha D_e/\lambda = 0.015(1 + 2.3\ D_e/L)\ (D2/D1)^{0.45}(v_a D_e/\nu)^{0.8}\ Pr^{1/3}.$$

This leads to a total $\alpha = \alpha_v + \alpha_a = 716$ W/(m^{2}K) and a heat transfer of $Q = \alpha\ A\ \Delta T =$ $= 6$ kW with an area A of 0.13 m^2 and temperature difference ΔT of 65K. These results indicate that it is possible to install suitable heat exchangers in the cooler and the boiler. The ideas have to be validated by experiments that determine the heat transfer coefficients.

Turbine layout

In this chapter the mechanical power conversion of the KDT is described. The reference thermal input is 20 kW, and basis is the T-S diagram in Figure 4 for p-xylene. The temperatures and pressures for the evaporator and the condenser are 573 K/2.06 MPa and 358 K/0.019 MPa respectively. The mass flow is calculated as 0.0287 kg/s and the exit velocity as 539 m/s taking into account a 15% loss due to entropy increase in the nozzle. The first stage of power conversion is given by the Laval nozzles embedded in the separator disk that rotates with the cylinder. As it is well known from a pure impulse turbine additional stages help to extract more energy contained in the fluid beam. One choice would be a traditional setup of rotor/guide wheel. Here we present a design based on a Ljungström wheel as the second stage. The power conversion is complemented by a by cooler blades that work as a boundary layer turbine. The setup is shown in Figure 5.

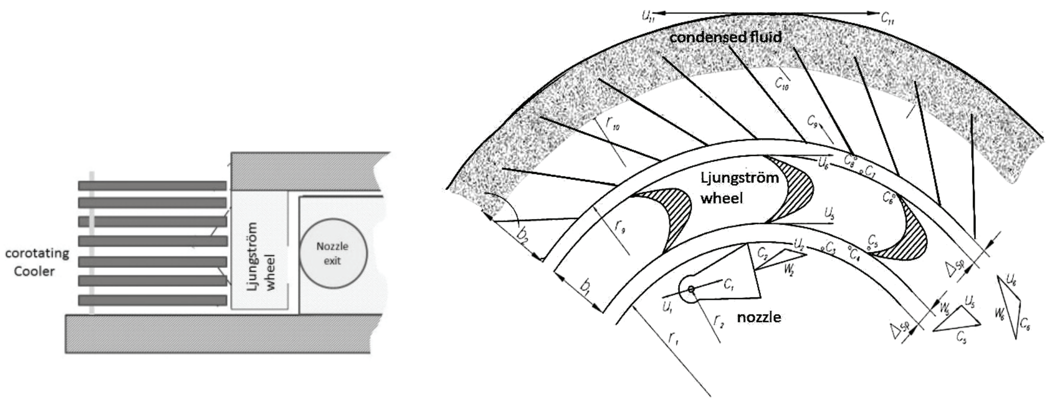


Figure 5. Sketch of the propulsion system: Nozzles, Ljungström wheel, cooler Propulsion system — top view schema — rotating nozzles, counterrotating Ljungström wheel and corotating condenser

The setup of the Ljungström wheel is given in Figure 5. The Ljungström wheel is counter rotating with respect to the KDT and transfers its power via a gear construction. Therefore high speeds of the wheel can be achieved. The friction of the Ljungström wheel is dominated by the term $(1 - \varepsilon)\ 0.75\ c_{rv}\ \rho\ n^3(2R)^4\ (l_b)$ where c_{rv} is an empirical constant [15].

Mechanical and thermodynamic losses

The main sources for losses in the system are the losses in the nozzles, wheel system with gaps in the cooler and the friction losses of the rotating cylinder and other mechanical parts like bearing, sealing, gear and rotating Ljungström wheel. The loss in the nozzle is taken into account by assuming an 85% isentropic efficiency in the calculation of the exit velocity of gas. We consider in more detail the losses in the Ljungström wheel, the cooler and the loss of the cylinder rotating in the ambient atmosphere of the rotating system. For the calculation of the loss in the wheel and the gap we use the approach which is described in MEI [16—23]. The losses of rotating cylinders due to friction have been well established [24] and are applied to the cylinder in Figure 3. It is possible to reduce the friction by encasing the rotating cylinder in a static casing [25]. Carrying out the detailed calculations for the output and the losses yields the results in Table 1.

Table 1

The summary of power outputs in the KDT. Efficiency for input of 20 kW thermal power into the KDT, reaches 11,1% ± 1.8% is at 991 rad/s with p-xylene and a Ljungström wheel with an angle velocity of ω = 2230 rad/s in the opposite direction

	Power [W]	remark
Nozzle	811	[10]
Ljungstrцm wheel	2711	[16, 17, 18, 19, 20, 21, 23]
Grid of Cooler	190	[29, 30]
Friction Ljungstrцm wheel	–276	[15]
Friction cylinder	–617	[24]
Friction Bearings and sealing	–199	estimation from www.skf.com
Friction of gear for Ljungstrцm wheel	–136	5% of power Ljungstrцm wheel
Mechanical Power	**2 484**	
Efficiency of Generator 92,5%		
Efficiency of DC/AC converter 96,5%		
Electrical Output	**2 217**	
Total Efficiency	**11,1%**	

According to this work the maximum power for a KDT will be achieved at about an angle velocity ω = 991 rad/s. Summarizing the efficiencies of the different parts of the turbine leads to Table 1. These considerations show that a total efficiency of 10% or more is possible. This result reflect the same trend as recent simulation studies of Kosowski et al. [26] which results also in approximately 10% efficiency for a small turbine with 25 KW thermal input power.

Summary and Outlook

The goal of this publication was to outline the basic ideas of the KDT. More detailed studies on condenser, evaporator and burner are necessary. The principles of the KDT are based on established research and technologies. The heat transfer in the boiler and the cooler will pose more severe challenges but the preliminary analysis indicates that there are no insurmountable difficulties. The total efficiency is estimated at 11% with an error of 1.8%

In the future also water as working fluid will be considered. The problem of high velocities and low mass flux can possibly be corrected by using the technique of steam mixtures [27]. This method has been used by KKK/Siemens [28] for A. However, it was used for mass flows that were larger by a factor of 20 as compared to this KDT.

Nomenclature

Symbol	Description	Symbol	Description
r_1	distance fluid surface to rotation axis in the cooler	L	characteristic length
r_2	radius upper cylinder (cooler)	η	dynamic.viscosity
ρ_1	density of liquid in the cooler	ρ	densitiy
T_1	temperature of liquid in the cooler	a	
p_1	vapor pressure in the cooler	Nu	Nusselt number
m_1	mass of liquid in the cooler	Re	Reynolds number
V_1	volume of liquid in the cooler	T	temperature
r_3	distance fluid surface to rotation axis in the boiler	δ	gap width in the annulli
r_4	radius lower cylinder (boiler)	α	heat transfer coefficient
ρ_2	density of liquid in the boiler	Pr	Prandtl number
T_2	temperature of liquid in the boiler	A	area
p_2	vapor pressure in the boiler	σ	Boltzmann constant
m_2	mass of liquid in the boiler	ε	emission coefficient
V_2	volume of liquid in the boiler	λ	thermal conductivity
ω	rotational frequency of KDT	Ta	Taylor number
D	diameter		indices
V	volume	v	volume
s	correction factor for gas	a	axial
p	pressure	f	fluid
ν	kienematic viscosity	w	wall
c_p	specific heat of the fluid	g	gas
l	length	t	turbulent
c_{rv}	empirical constatn for loss calclation	l	laminar
n	rotational frequency	e	equivalent
R	radius	b	blade

This analysis gives strong indication that the KDT can be used as the core for a micro CHP and as cheap power generator for small solar power plants at off-grid locations. The economical and environmental impact of the KDT could be severe. In Germany alone app. 700.000 residential heatings are replaced every year. These will all become targets for a CHP based on an efficient and affordable KDT. In addition the push towards more efficient use of energy is driven by the requirement to drastically reduce the CO_2 output further encouraging the deployment of co-generation plants. In industrialized countries the existing grid will me more and more complemented by decentralized energy production. In many countries electrical grids do not exist, but solar radiation is available. For those places the KDT offers the means of cheap production of electrical energy to run computer and communication equipment.

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ИНТЕГРИРОВАННАЯ ТЕПЛОВАЯ СТАНЦИЯ С КОМПАКТНОЙ ПАРОВОЙ ТУРБИНОЙ, РАБОТАЮЩАЯ НА НИЗКОКИПЯЩЕМ РАБОЧЕМ ТЕЛЕ

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При когенерации происходит совместное производство электрической и тепловой энергии. В настоящее время когенерация распространена при средних и больших мощностях (> 100 кВт). Компактная паровая турбина (KDT) относится к машинам с электрической мощность около 2 кВт. KDT является высокоинтегрированной тепловой машиной, которая в состоянии использовать различные источники тепла. Простая конструкция KDT предполагает экономически выгодную выработку тепловой и электрической энергии. В статье приводятся тепловые, газодинамические и прочностные расчеты KDT.

Ключевые слова: интегрированная тепловая станция, турбина, термодинамические потери, когенерация.