

# **Long-term experience report on the use of screw expanders in commercial small power plants named IC60 based on the Organic Rankine Cycle (ORC)**

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## **Abstract**

GMK manufactures the ORC plants named IC60 and presented in [1] in a small series production at their headquarters located near Rostock and reports now about the experience in the commercial operation. Strongly fluctuating heat sources and heat sinks means high requirements on the ORC plant in everyday use. The used oil-injected screw expander in connection with the shell and tube heat exchangers has been proven to be as extremely robust and reliable in every way. Currently two expander sizes operate on different rotational speeds and pressures in the IC60 ORC plants from GMK. Over the last few years GMK has developed and implemented different operating strategies, to maximize both the electrical power output and the economics at the same time.

## **1. IC60 ORC-plant**

GMK has advanced their ORC system after the presentation of the prototype plant concept at the last VDI screw machine conference in 2010 [1]. They managed to make the system components even more efficient and merge them into a compact arrangement on a gen-set base frame with the footprint of approx. 1.1 x 4.5 m incl. switchgear (see Fig. 1). Heart of the system is an oil injected rotary twin screw expander which is based on a multi-modified series refrigerant compressor. Important key components such as evaporator, oil separator and expansion tank are produced in Chemnitz within their own Company Group Germania and will be supplied to GMK. The specially developed operating program ensures fully automatic and efficient operation only with surplus waste heat. The flexible PLC control allows it to react on customer requirements which differ from the series.

A 20 feet special container housing provides the necessary protection from the weather as well as ventilation and sound insulation of the ORC unit in the field. At the same time the container is the base for the air cooled condenser, which V-shape and fan technology combines large heat transfer surfaces and superior efficiency on the existing footprint.

The company GMK manufactures the ORC plants designated as IC60 in small series at the location near Rostock and reports in the following about the technical and economic experience in commercial operation.



Fig. 1: IC60 ORC unit (Series No.1)

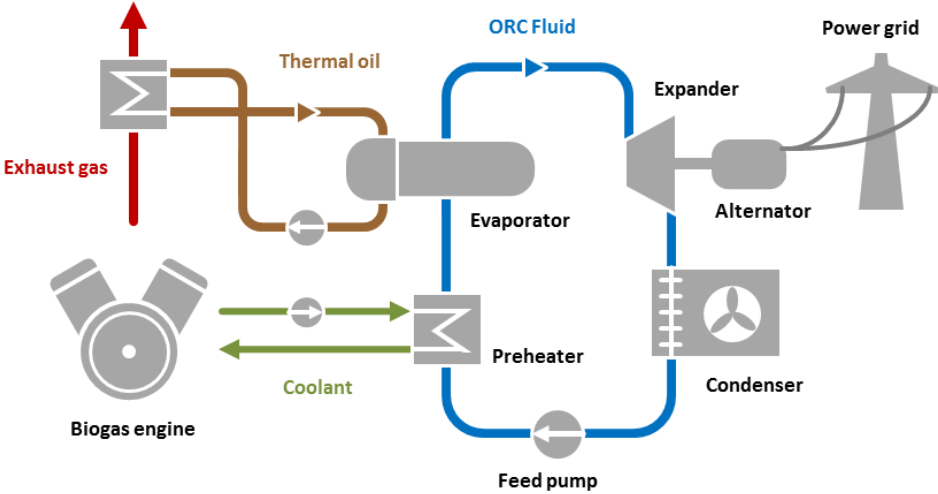


Fig. 2: Rough process scheme of an integrated IC60 ORC plant

## 2. Applications

There are currently more than 15 units of the type IC60 in the field. The customers are mostly owners of biogas plants, who want to turn their waste heat, produced during the power generation with stationary gas engines, into electricity. These heat sources have, inter

alia, the peculiarity that the gas engines in spite of their good electrical efficiency produce high amounts of waste heat that cannot be used sufficiently in the rural locations on site. The IC60 ORC plant is available in several configurations for gas engine sizes from 400 up to 700 kWe power and reaches itself gross power outputs up to 60 kWe. An IC60 ORC plant therefore leads to a significant increase of the overall efficiency of the biogas plant (see Fig. 4), so that the net power output usually can cover the complete auxiliary power demand of the biogas plant for consumers, such as compressors, pumps, agitators, etc.



Fig. 3: Biogas plant in Pomerania (Northeast Germany) with a 500 kWe gas engine with an exhaust gas heat exchanger and an integrated IC60 ORC unit with an air cooled condenser.

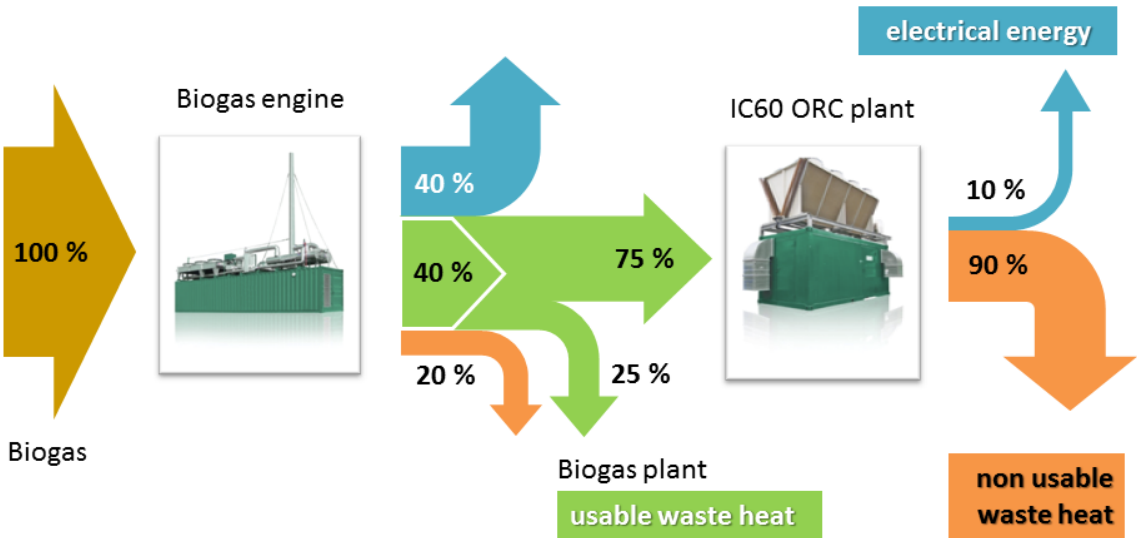


Fig. 4: Rough Energy flow diagram of a biogas engine with attached IC60 ORC plant.

### 3. Characteristic of heat sources and heat sinks

On biogas plants that are recommended for retrofitting with an IC60 ORC plant, there are typically only a few consumers with low heat demand. Usually the largest heat load is represented by the fermenter. Depending on the season, execution and operation of this component, the quality and quantity of heat consumption for process heating strongly deviates. Fig. 5 illustrates the high fluctuations in the heat supply and the electrical power output as a result of intensive cyclic fermenter heating. For an optimum yield and efficiency it is advisable to generate a steady heat supply on the one hand, but also a steady demand of electricity for the biogas plant consumer on the other side. Some customers have already recognized this and optimized their biogas plant to that effect. One reason for much higher variations in the heat capacity are gas engines which operate at variable load levels, due to the requirements of the grid operator or the electricity market. In case of an ORC plant designed for a standard operation with an engine operating at full load, this inevitably leads to frequent ORC partial load operation with low vapor pressures and reduced expander efficiency.

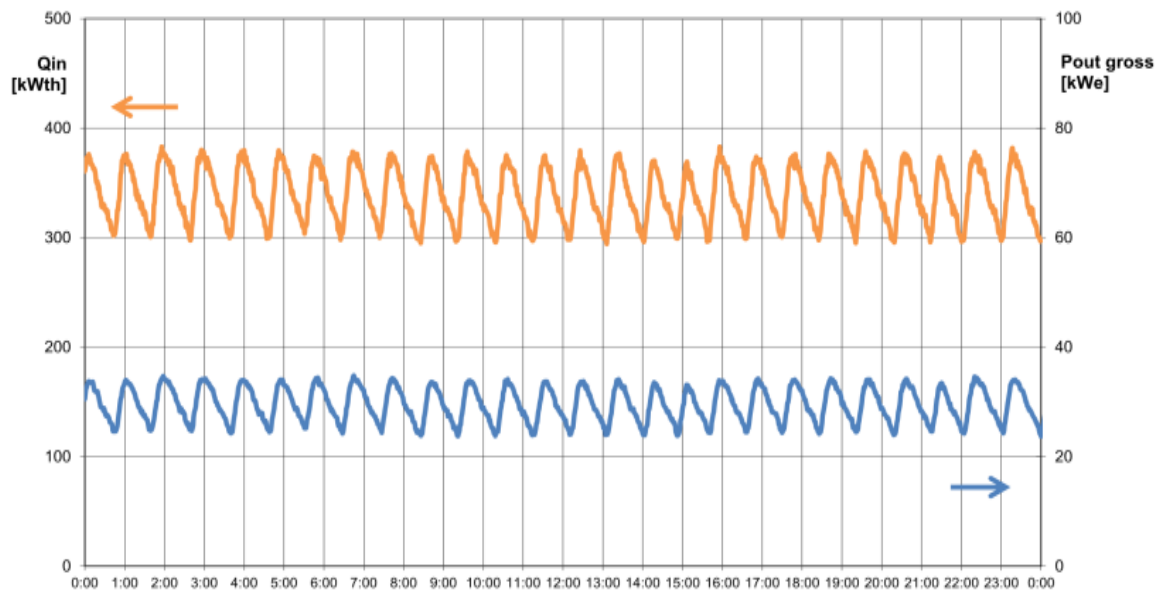


Fig. 5: Intraday plot of the supplied heat power input ( $Q_{in}$ ) and the electrical gross power of an IC60 ORC plant ( $P_{out\ gross}$ ) at a biogas plant with cyclic fermenter heating on a cold day

Another significant influence on the operating conditions of air cooled plants is the heat sink. For the electrical power output of an Organic Rankine Cycle it is advantageous, that the condensation occurs at the lowest possible temperature level as possible. In addition to the

increasing efficiency of the cycle, the lower condensation temperature cause that more engine coolant heat can be transferred to the ORC fluid and converted into electricity. An efficiency control ensures that the electrical fan consumption does not disproportionate increases at the same time. Inevitably, however, the condensation pressure rises with the air temperature on warmer days. The result is a decreasing efficiency of the cycle and the expander itself. Due to the high condensation temperatures less heat can also be taken by the ORC fluid. The influence on the gross electrical power output of an IC60 plant that is optimized for the cold season and a high coolant heat input, is represented in Fig. 6.

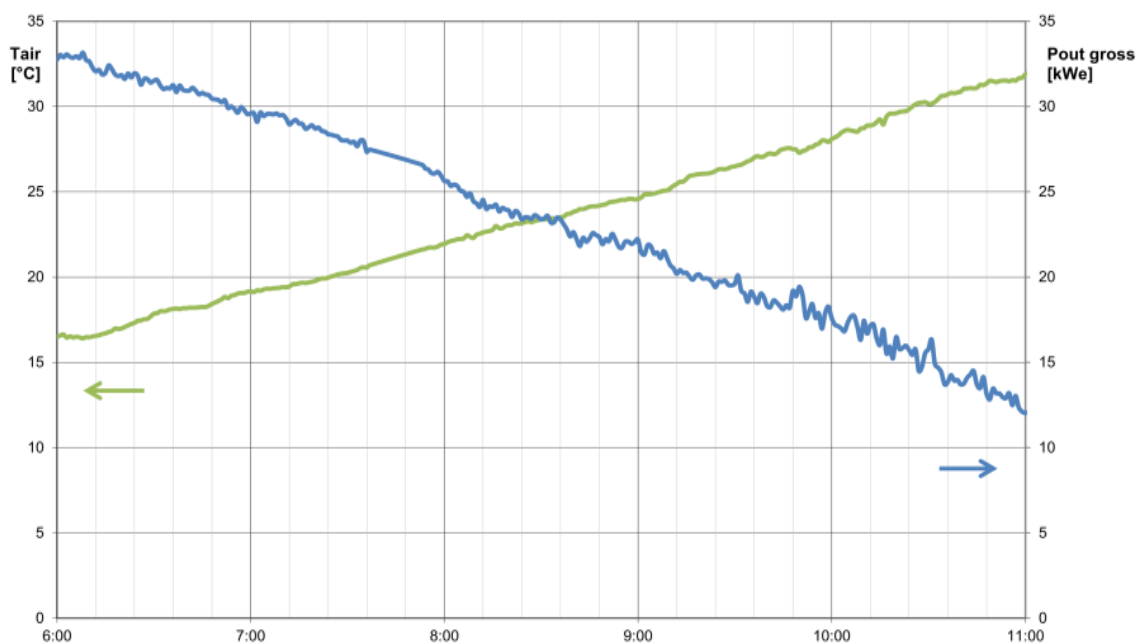


Fig. 6: Intraday plot of the cooling air temperature ( $T_{air}$ ) and the electrical gross power output ( $P_{out\ gross}$ ) of an IC60 ORC plant which is running on low vapor pressure level in the morning of a warm day

#### 4. Operating Strategies

Especially on warm locations with a high annual average air temperature ( $\rightarrow$  high outlet vapor pressure  $p_2$ ) or a lower heat input, e.g. through frequent part-load operation ( $\rightarrow$  low inlet vapor pressure  $p_1$ ), the experience has shown that the expander will be operating with an inefficient Over-expansion for large periods of the year. In this case the external plant pressure ratio is smaller than the internal expander pressure ratio. The effective usable expansion work is reduced, due to the fact that the expanded ORC fluid has to be pushed out against the higher condenser pressure. (See areas in Fig. 7)

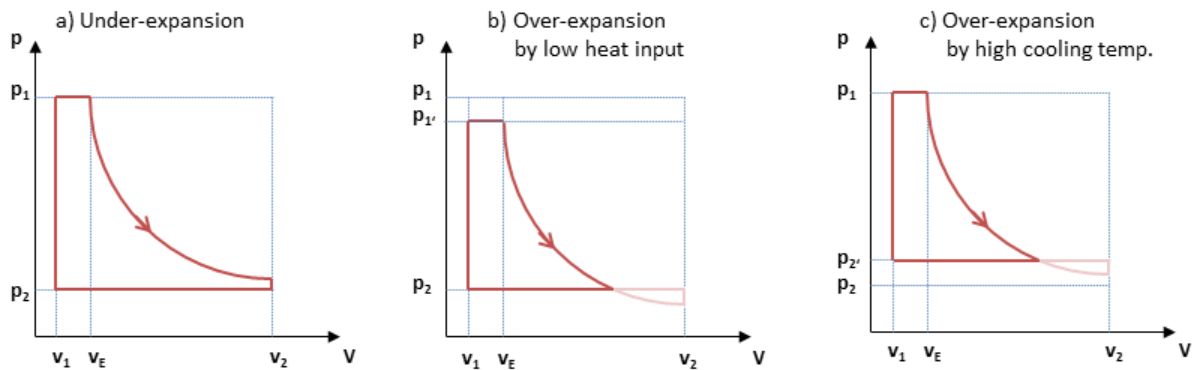


Fig. 7: Ideal p-V-Diagram of the expander for the 3 most common operational cases of IC60 ORC unit

On the basis of the high temperature thermal oil circuit in combination with the robust shell and tube evaporators, the IC60 ORC units are able to generate a wide vapor-pressure spectrum. Due to a drastic reduction of the expander shaft speed a significant increase of the inlet pressure could be achieved and therewith an operation in Under-expansion could be ensured for almost the whole year. In consequence of the shaft speed reduction and the prevailing higher pressure difference, there was the significant increase in volumetric efficiency, as expected. But contrary to the expectations the effective isentropic grade of the expander decreased only a few percentage points.

As an alternative to the rotational speed reduction, a smaller expander series was also tested and operated with identical internal volume ratio. But the combination of higher shaft speeds and a smaller inlet cross section, however, resulted in significantly lower volumetric efficiency, which can be clearly attributed to the filling factor according to [2].

For locations with predominantly volatile heat input, GMK now offer its clients a variable speed solution. A feed-in capable frequency converter adjusts the shaft speed, so that the expander always operates near the aligned mode in which the external system pressure ratio is matched to the internal volume ratio of the expander. This concept is able to generate the maximum alternator power for an existing plant configuration under all circumstances. The disadvantages of the frequency converter are the inherent conversion losses, the expanded plant complexity and the additional investment. The confrontation with a solid designed fixed-speed unit revealed that the seasonal efficiency gains of this concept were largely compensated by the full-year losses of the permanent frequency conversion. From the perspective of the authors, therefore this option is not effective for locations without large fluctuations in heat source and heat sink.

## 5. Conclusion

The Basics for the maximum power output at one location is the knowledge about the level and variation of heat range in the future, while also taking into account the climatic and installation conditions on site, such as the geodetic height. That is the only way to define an optimum design point for an ORC plant.

According to equation (1) the mechanical shaft power of the expander  $P_{\text{mech}}$  results from the isentropic grade of the expander ( $\eta_{\text{is,eff}}$ ), the ideal thermodynamic cycle efficiency  $\eta_{\text{ORC,ideal}}$  and the supplied heat flux ( $Q_{\text{in}}$ ). As described in the previous chapter, for a maximum heat input  $Q_{\text{in}}$ , a possible "cold" cycle with a low spread is advantageous. Due to the low condensation temperatures and the high ORC fluid mass flow, the IC60 ORC unit can receive a lot of engine coolant heat. In contrast to that a possible high cycle spread (delta temperature) has a positive effect on  $\eta_{\text{ORC,ideal}}$ . The optimal  $\eta_{\text{is,eff}}$  arises when the system pressure ratio that is equal or higher than the internal pressure or volume ratio. For an optimum electrical performance yield the product of these 3 factors should reach a maximum value, which in practice usually was observed at moderate Under-expansion in the cold half of the year.

$$P_{\text{mech}} = \eta_{\text{is,eff}} \times \eta_{\text{ORC,ideal}} \times Q_{\text{in}} \quad (1)$$

$$\eta_{\text{is,eff}} = \eta_{\text{is,in}} \times \eta_{\text{mech}} \quad (2)$$

Regarding the shaft speed reduction of the bigger machine, the effect of only minor losses in the expander efficiency in spite of the significant increase in volumetric efficiency can be explained by formula (2). After this, the effective isentropic grade  $\eta_{\text{is,eff}}$  consist of the inner isentropic grade of the expansion  $\eta_{\text{is,in}}$  and the mechanical efficiency of the machine  $\eta_{\text{mech}}$ . Due to the increase in the volumetric efficiency the inner isentropic grade decreases, but can be partially compensated by the now prevailing higher pressure ratio as described in [2]. Furthermore a shaft speed reduction has a positive effect on the friction losses within the machine. The relative friction loss decreases, which manifests in a higher mechanical efficiency. Together with the increased thermodynamic cycle efficiency, due to the higher pressure ratio, this expander configuration has become the standard solution for locations with low to medium amount of heat. A notable side effect of the high volumetric efficiency is that the efficiency drop of the expander at Over-expansion is significantly weaker than in operation with high speeds.

On the example of the small expander it can be seen that inside a machine with unfavorable ratio of inlet cross section to swept volume, the throttle effects during the filling process become a higher significance than the bypass losses over the machine gaps. The effective isentropic grades of the small expander are consistently lower than those of the large series at comparable operating conditions. In order to compensate the pressure drop through inlet throttling the affected IC60 ORC unit currently operating with a high cycle (temperature) spreading with the disadvantages mentioned above.

## **6. Review and Challenges for the Future**

With the IC60 ORC plants in use, GMK now has an experience of almost 200,000 hours of operation. During this time, important insights were gained, which allows a good prediction of the actual produced heat at a focused location, which often deviates significantly from the theoretical values. The recorded data from the various IC60 installations with their expander operating on different conditions provided the authors with a valuable basis for performance calculations and implemented optimizations. Due to validated cycle simulations GMK is now in a position to create a secure power prediction for their customers and can avoid high performance safety deductions. With the addition of climatic data sets and a comparison with existing systems, a good prediction of the expected annual electrical work is also possible, which is essential for the economic evaluation of the owner or operator.

Despite the sometimes harsh operating conditions with regularly load drops as a result of power outages or permanently cyclic modes of operation, the total system components have proved to be extremely robust and reliable in every way. In particular, the expander shows no significant signs of wear. Continuous improvements regarding efficiency and reliability allow regular plant availability of more than 100% - based on the uptime of the biogas engine.

Against the background of decreasing subsidies for renewable energy and increasing awareness of resource efficiency, the company's focus is clearly on the development efficient and cost effective ORC plants.

### **Literatur**

- [1] Drescher, U. et. al., Stromerzeugung mit einem Schraubenexpander in einem Organic Rankine Cycle, VDI Berichte 2101, 2010
- [2] Zellermann, R., Optimierung von Schraubenmotoren mit Flüssigkeitseinspritzung, VDI Fortschrittsbericht Nr. 357, 1997