Direct solar air heating inside small-scale linear Fresnel collector assisted by a turbocharger: Experimental characterization

Antonio Famiglietti*, Antonio Lecuona

Universidad Carlos III de Madrid, Departamento de Ingeniería Térmica y de Fluidos, Avda. de la Universidad 30, 28911 Leganés, Madrid, Spain

A R T I C L E   I N F O

Keywords:
Linear Fresnel collector
Solar air heater
Turbocharger
Solar Brayton cycle
Solar heat for industrial processes
Solar drying

A B S T R A C T

In recent years many efforts are being made to replace fossil fuel with renewable energy sources in the industry, both for electricity and heat production. Several high-energy demanding processes use hot air as heat and moisture carrier in the low and medium temperature range, for which solar air heaters offer great potential. Among solar energy technologies, Linear Fresnel collectors can provide heat in the medium temperature range at a decreasing cost. Direct air heating inside Linear Fresnel collector avoids the use of a liquid heat transfer fluid and the heat exchanger, reducing installation and maintenance costs, as well as residues. In this innovative technology, an automotive turbocharger reduces auxiliary energy consumption for air pumping through the solar collector, which would be unsustainable at medium and large scale. It configures an original open-to-atmosphere Brayton cycle with null power efficiency. In this study, an experimental investigation is carried out on the first small-scale prototype solar field of 79.2 m² area. It allows characterizing the thermal and mechanical behavior of the turbocharger, besides tuning and validating the numerical model implemented. Special attention is given to the diabatic behavior of both turbomachines and to establish the limiting efficiency of the turbocharger as forefront research.

1. Introduction

The industrial sector takes a significant share of the global energy demand, requiring both electricity and heat. Industrial heat is mostly provided from fossil fuels and frequently from electricity, which is nowadays produced partially or from renewable sources. The adoption of solar energy in industry as a clean source has a great potential for decarbonization, as well as for pollutant emission reduction and environmental sustainability.

Sharma et al. [1] reviewed the solar potential in industrial heating for several countries, reporting mainly low to medium temperature processes for several industrial sectors: food and beverage, textile, pulp and paper, chemical, casting industry, cement, rubber, and plastic manufacturing. Solar Heat for Industrial Processes (SHIP) state of art and potential are reviewed by [2]. According to [3] and [4], around half of the global industrial heat demand is in the low and medium temperature range (below 400 °C). As reported by [5], about 39% of industrial energy demand in Europe is heat below 500 °C, 21% between 100 and 200 °C, 9% between 200 and 500 °C, and 9% below 100 °C. According to the required temperature, several technologies are available for solar industrial heating, e.g. [6]. For low-temperature applications (below 100 °C), Flat Plate Collectors (FPCs) and Evacuated Tube collectors (ETCs) are the most common solution. Solar Air Collectors (SACs) can be an alternative where low-temperature hot air is needed. Medium temperature heat, up to 400 °C, can be provided by using concentrating linear technologies, such as Parabolic Trough Collectors (PTCs) and Linear Fresnel Collectors (LFCs). Higher temperatures can be reached with central receivers and solar towers.

According to Task 49 IEA report [7], 635 SHIP plants are in operation around the world. The industrial sector with more installed SHIP is food and beverage, followed by textile and pharmaceutical. The majority use flat-plate collectors followed by evacuated tube collectors. 88% of the plants use non-concentrating technologies. Existing SHIP plants are reported in the SHIP database [8] developed within the IEA SHC Task 49/ SolarPACES, as well as by [1] and [2].

Among the large energy consumption in industry, hot air production is relevant. Several high-energy demanding processes require hot air as a heat transfer medium, either in the low and in the medium temperature range. Drying is one widely used technique requiring hot air, common to different industrial sectors, including food and beverage, wastewater treatment, mining, manufacturing, agroindustry, among others, e.g.
### Nomenclature

**Latin**
- **ac**: Auxiliary compressor
- **A_m**: Active LFC area \([m^2]\)
- **c**: Compressor
- **c_p**: Air constant \(p\) specific heat capacity \([J \cdot kg^{-1} \cdot °C^{-1}]\)
- **c_e**: Turbine mass flow fitting function coefficient \([kg \cdot s^{-1}]\)
- **c_ec1**: Fitting function coefficient \([W \cdot °C^{-1}]\)
- **c_ec2**: Fitting function coefficient \([-]\)
- **D**: Inner diameter of the receiver tube \([m]\)
- **e**: Turbine
- **f**: Darcy friction factor \([-]\)
- **f_D**: Darcy friction factor \([-]\)
- **H**: Frequency \([Hz]\)
- **H_r**: Receiver height \([m]\)
- **h_con**: Convection heat transfer coefficient \([W \cdot m^{-2} \cdot °C^{-1}]\)
- **h_rad**: Radiation heat transfer coefficient \([W \cdot m^{-2} \cdot °C^{-1}]\)
- **IC**: Interpolation ratio \([-]\)
- **J_T**: Turbocharger polar inertia \([kg \cdot m^2]\)
- **K**: Localized pressure drop coefficient \([-]\)
- **k_e**: Turbine mass flow fitting function coefficient \([-]\)
- **k_i**: Compressor mass flow fitting function parameters \([-]\)
- **L**: Length \([m]\)
- **L_m**: LFC length \([m]\)
- **L_r**: Receiver overall length \([m]\)
- **L_n**: Connection pipe length \([m]\)
- **m_a**: Air mass flow rate \([kg \cdot s^{-1}]\)
- **Ma**: Mach number \([-]\)
- **n**: Connection
- **n_c**: LFC in series \([-]\)
- **n_T**: Turbocharger speed \([rpm]\)
- **n_h**: Number of mirrors
- **p**: Post-heating
- **p_h**: Post-heating
- **r**: Receiver
- **ref**: Reference value
- **s**: Isoentropic
- **t**: Total or stagnation variable
- **u**: Useful

**Subscripts**
- **a**: Air
- **ac**: Auxiliary compressor
- **af**: After
- **amb**: Ambient
- **be**: Before
- **car**: Corrected
- **c**: Compressor
- **c→amb**: Compressor to ambient
- **e**: Turbine
- **e→c**: Turbine to compressor
- **e→amb**: Turbine to ambient
- **ex**: External
- **f**: Frictional
- **in**: Inlet
- **k**: Kinetic
- **m**: LFC module, mirror
- **n**: Connection pipe
- **net**: Net
- **opt**: Optimum
- **ou**: Outlet
- **ph**: Post-heating
- **r**: Receiver
- **ref**: Reference value
- **s**: Isoentropic
- **t**: Total or stagnation variable
- **u**: Useful

**Superscripts**
- **adi**: Adiabatic
- **cal**: Calculated value
- **dia**: Diabatic
- **exp**: Experimental value
- **min**: Minimum
- **max**: Maximum
- ***: Corrected with parasitic pressure losses

**Acronyms**
- **AC**: Air Collector
- **ETC**: Evacuated Tube Collector
- **FPC**: Flat Plate Collector
- **HTF**: Heat transfer fluid
- **HX**: Heat exchanger
- **LFC**: Linear Fresnel Collector
- **PTC**: Parabolic Trough Collector
- **SAH**: Solar Air Heater
- **SHIP**: Solar Heat for Industrial Processes
- **T-SAH**: Turbo-assisted Solar Air Heater

**Others**
- **[]**: Functional dependence
Existing Solar Air Heaters (SAHs) can directly heat air using solar energy. The variety of developed devices work with natural and forced convection flow, single or double-pass configuration, finned surface, and even using turbulators for enhancing heat transfer rate, e.g., [12]. They rarely provide hot air above 100 °C for what they are a suitable solution for low-temperature applications.

Direct solar air heating above 150 °C lacks examples, although the potential applications are widespread. Rare examples are reported in the literature regarding solar energy usage for indirect air heating, using concentrating collectors. For example, [8] reports a 105 m² plant of Parabolic Trough Collectors (PTCs) heating air for drying finished products in Portugal. [13] described a 132 m² LFCs field providing hot air for paint curing for the automotive industry in Germany. LFCs are also used to provide hot air for indirect evaporative cooling [14]. All the mentioned examples use a primary Heat Transfer Fluid HTF (thermal oil or water) to evacuate heat from the collectors and HTF/air Heat exchangerHX to heat process air as a secondary HTF. Using thermal oil implies non-negligible installation, replacement, and disposal costs, present degradation at high temperature, leakages problems related to environmental pollution, and fire risk. Water implies the drawbacks of boiling and two-phase flow unless pressurization is achieved, increasing risks related to leakages, besides freezing during the night, and corrosion. A primary pump is needed for circulating the HTF. Besides its relevant installation cost, the use of an HTF/air heat exchanger introduces an exergy loss, needs an additional blower for the airflow, and requires space availability.

Although still uncommon, the direct solar air heating inside concentrating collectors, either with PTCs or LFCs, would reduce the complexity and the cost of a solar facility, avoiding the need for a primary HTF as well as an HTF/airHX. As for a non-concentrating solar air heater, an Open-to-Atmosphere OA circuit could be implemented with the advantage of higher delivery temperature, achievable thanks to the high thermal efficiency of linear concentrating collectors. Experimental information about this solar energy application is not available in the open literature.

A seminal work [15] theoretically investigated the direct air heating inside Linear Concentrating Collectors from a theoretical point of view. Despite using air as HTF is uncommon due to the unfavorable thermal performance of air compared with conventional HTFs (thermal oil, water, steam), the authors suggested that direct air heating is an interesting solution for industrial applications that use hot air as a process fluid. They pointed out that direct air heating at atmospheric pressure is viable but limited to small collector rows and moderate solar heat flux (low optical concentration), since the external auxiliary power required for air pumping grows with mass flow rate used, hence with collector aperture and length. According to this same work [15], to directly heat air to 350 °C, an LFC with 5 m of aperture, concentrating a direct normal solar irradiance of 1000 W m⁻² with an optical efficiency of 0.5 on the external perimeter of a standard vacuum receiver tube of 0.07 m diameter, thus with a concentration ratio of 22.7 and heat flux 12 kW m⁻², requires an isentropic pumping power in the order of 10% of net solar power gain for a collector length of 30 m, growing to 20% for a collector length of 40 m, more than 50% for a length of 50 m.

To minimize these drawbacks they proposed an OA Brayton cycle configuration, increasing the inlet air pressure up to 2–3 bar using a turbocompressor and recovering the compressing power through a turbine installed at the receiver outlet, as in Fig. 1. By increasing air density inside the receiver, this innovative layout allows minimizing the stagnation pressure drop across the solar field, reducing the mean flow velocity for a constant mass flow rate. The turbine provides the mechanical work needed for compression and pumping, avoiding any external auxiliary energy consumption. Providing or extracting mechanical power at the shaft is out of the scope of the system, for what an off-the-shelf automotive turbocharger can be used, aiming at low cost, where a compressor and a turbine are joined in a compact device. Manufactured by millions every year all around the world by the automotive industry for boosting internal combustion engines performances, the turbocharger is extremely cheap, reliable, and robust, offering small size and low weight. While the inlet turbine temperature does not exceed 550 °C according to the thermal limit of vacuum receivers, at the turbine exit the Turbo-assisted Solar Air Heater T-SAH there proposed can deliver air between 300 °C and 400 °C, a temperature suitable for transportation to the point of consumption and for driving a thermal air-based industrial process. This way replacing or integrating the existing conventional air heating facility. For a well-designed system, an auxiliary compressor, Fig. 1, is needed only for the starting transient, or control purposes, with negligible auxiliary energy consumption.

A further study [16] applied the T-SAH concept to a medium-scale facility with 633.6 m² solar field using LFCs with a lateral aperture of 5 m arranged into 4 parallel loops with an irradiated length of 30 m. They implemented a detailed numerical model of the turbocompressor and the solar field of LFCs to evaluate the daily and yearly performance of T-SAH under a typical meteorological year of Madrid (Spain). The results confirmed its technical viability, indicating that the proposed T-SAH can provide 330 MWh of heat per year, delivering hot air in the range of 300 °C to 400 °C, while working around 2000 h, without the aid of an auxiliary compressor, thus avoiding external energy consumption for pumping. The air temperature reached inside the receiver tube does not overcome 550 °C while the maximum receiver temperature is below its thermal limit, 600 °C.

In the present study, an experimental analysis of the T-SAH technology is carried out on an original laboratory prototype, described in Section 2, based on a first-generation small-scale commercial LFC field and a commercial turbocompressor. A turbocompressor model is implemented in Section 3, combining conventional and original techniques. It is based on the performance maps provided by the manufacturers as well as on complementary theoretical assumptions. The developed modeling
procedure was applied to the turbocharger understudy and holds general validity for its application to other devices. Section 4 evaluates and discusses the experimental results. Owing to the specific issues of the T-SAH concept, this section also analyzes the undesired heat transfer phenomena affecting turbocharger performance, as its efficiency is of paramount importance. The optical and thermal efficiency of LFCs are not analyzed in this work. Within the present layout, the experimental campaign performed under different operating conditions allowed identifying the relevant design and operating parameters as well as accurately characterizing the turbocharger behavior. Experimental data obtained allowed extending, verifying, and turning the numerical model as well as quantifying the heat transfer phenomena affecting the diabatic turbocharger. A detailed discussion of the results highlights the improvements possible in scaling up the technology and streamlining the design for industrial applications. Section 5 summarizes the relevant conclusions.

2. Experimental setup

Following the T-SAH concept proposed by [15], an experimental setup was installed at Carlos III University of Madrid (Spain), in the city of Leganés, Fig. 2. A small-scale solar installation was built using three linear Fresnel collectors in series, Fig. 4a, Fig. 4b, manufactured by [17], whose main technical parameters reports Table 1. The primary reflector is composed of 10 mirrors of lateral aperture \( w_m = 0.50 \text{ m} \), individually motorized for tracking the sun, according to an astronomical algorithm, Fig. 3. A secondary reflector having a trapezoidal shape refocuses on the receiver the part of the sun rays missing it at the first reflection. This allows increasing the optical efficiency and homogenizing the circumferential distribution of concentrated irradiance on the receiver perimeter. Five evacuated solar tubes, with an external diameter \( D_{ex} = 70 \text{ mm} \) and a maximum allowable wall temperature of \( 600 \degree C \), are used as the receiver. They are manufactured by [18] (model HCEOI-12), being developed for concentrating solar plants (CSPs) operating with thermal oil as HTF. The stainless-steel absorber tube is covered by a selective coating able to reduce thermal radiation to the ambient. The selective coating based on multilayer structure has high absorptance in the solar wavelength and low infrared emissivity at the operating receiver temperature. The absorber is embedded into a coaxial cover of borosilicate glass that is coated with an anti-reflective layer. The vacuum in the annulus minimizes convective thermal losses. Thermal longitudinal expansion of stainless steel tube is compensated by extensible bellows, which are welded on the absorber tube and on the glass jacked to keep vacuum inside the annulus. Tab. 1 reports the main receiver technical information taken from the manufacturer datasheet.

The automotive turbocharger model GT1544 by [19] was chosen...
among the small-size models available on the market, Fig. 4d. The compressor and the turbine have respectively wheels diameter of $D_c = 43.9$ mm and $D_e = 41$ mm; TRIM, as the ratio between smallest and highest wheel area, is given as 56% and 58% respectively. An $A/R$, as the ratio between the interior area of the compressor/turbine and the corresponding housing radio, is given as 33% and 34% respectively. The turbine is originally equipped with a wastegate valve, which has been closed permanently. Shaft standard plain bearing lubrication is ensured by a continuous supply of pressurized oil via a dedicated pump. More efficient ball bearing and even oil-free technology are commercially available for large-size turbochargers.

An auxiliary compressor unit $ac$ is mounted in series with the turbocompressor, formed by two in parallel electrical side-channel blowers (Elektor SD22 FU 80/1,1), Fig. 4c. They are equipped with ball bearings and do not need lubrication. Their asynchronous squirrel-cage electrical motors (rated power 1.1 kW at 50 Hz) are fed by a variable frequency converter (Lense 8200 vector).

An experimental air post-heating unit $ph$ has been added to the setup to increase the inlet turbine temperature. It is used only for experimental purposes and would not be implemented in an industrial scale plant, where the use of electricity for air heating is not coherent. In the experimental setup, the experimental post-heating unit goal is to

### Table 1

<table>
<thead>
<tr>
<th>LFC module, solar field, and receiver parameters.</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>LFC module active length $L_m$</td>
<td>5.28m</td>
</tr>
<tr>
<td>Mirrors per module $n_m$</td>
<td>10</td>
</tr>
<tr>
<td>Mirror aperture width $w_m$</td>
<td>0.50m</td>
</tr>
<tr>
<td>LFC aperture width $W_m = w_m n_m$</td>
<td>5.00m</td>
</tr>
<tr>
<td>Height receiver $H_m$</td>
<td>2.72m</td>
</tr>
<tr>
<td>LFC active area $A_m = L_m W_m$</td>
<td>26.40m²</td>
</tr>
<tr>
<td>Normal optical efficiency $\eta_{opt}$</td>
<td>0.632</td>
</tr>
<tr>
<td>LFC in series $n_s$</td>
<td>3</td>
</tr>
<tr>
<td>Overall active area $A_{tot} = n_s A_m$</td>
<td>79.2m²</td>
</tr>
<tr>
<td>Overall receiver length $L_r$</td>
<td>20.65m</td>
</tr>
<tr>
<td>Receiver tube length $L_t$</td>
<td>4.06m</td>
</tr>
<tr>
<td>Receiver internal diameter $D_i$</td>
<td>0.066m</td>
</tr>
<tr>
<td>Receiver external diameter $D_{ex}$</td>
<td>0.070m</td>
</tr>
<tr>
<td>Selective coating solar absorbance</td>
<td>95.8%</td>
</tr>
<tr>
<td>Selective coating thermal emissivity (400 °C)</td>
<td>8.2%</td>
</tr>
<tr>
<td>Selective coating maximum operating temperature</td>
<td>580°C</td>
</tr>
<tr>
<td>Selective coating maximum allowable temperature</td>
<td>600°C</td>
</tr>
<tr>
<td>Glass cover external diameter</td>
<td>0.125m</td>
</tr>
<tr>
<td>Glass cover thickness</td>
<td>0.003m</td>
</tr>
<tr>
<td>Average solar transmittance</td>
<td>96.4%</td>
</tr>
<tr>
<td>Annulus pressure</td>
<td>10⁻⁵mbar</td>
</tr>
<tr>
<td>Concentration ratio $C = W_m/(D_{ex} \pi)$</td>
<td>22.74</td>
</tr>
<tr>
<td>Connection pipe $n_1$ length $l_{con}$</td>
<td>26m</td>
</tr>
<tr>
<td>Connection pipe $n_2$ length $l_{con}$</td>
<td>3m</td>
</tr>
<tr>
<td>Connection pipe internal diameter $D_{con}$</td>
<td>0.08m</td>
</tr>
<tr>
<td>Orientation (0 = South) $\gamma$</td>
<td>54.3degW</td>
</tr>
<tr>
<td>Latitude $\Phi_{loc}$</td>
<td>40.165degN</td>
</tr>
<tr>
<td>Longitude $\lambda_{loc}$</td>
<td>3.704degW</td>
</tr>
</tbody>
</table>
provide additional thermal power as would be done by an extra collector row length in series with the installed collectors. The advantage of using an electrical heater for experimental purposes is the lower cost compared with an extra collector length and the possibility of easy and precise control of power delivery and temperature under the varying solar source. Moreover, the experimental post-heating unit allows increasing the air temperature above the thermal limit of the present solar source. Moreover, the experimental post-heating unit allows increasing the air temperature above the thermal limit of the present receiver tubes without any risk of their damage. This way the post-heating power allows to test the turbocharger behavior under a wider range of operating conditions than the sole operation with the solar collector only. Moreover, it allows compensating the low turbine efficiency, because of its small size. It increases the overall thermal power supplied to the air by adding up to 8 kW, which ranges between 15 and 20 kW. The experimental post-heating unit is equipped with 8 electrical resistances capable of supplying heat to the fluid up to 650 °C. They are made of sheathed heating wire of 2.0 m length having an electrical resistance of 50.0 Ω. The electrical heaters are mounted on a high-temperature resistant steel structure capable of improving heat transfer to the air and protecting the resistances from the risk of overheating. At the same time, this structure minimizes the additional pressure drop introduced. It is equipped with a suitable power supply and control unit. The electrical heaters and supporting structure were installed inside the air circuit between the receiver and the turbine, as in Fig. 2.

Thick wall stainless steel pipes connecting the turbocharger with the solar receiver, n1 and n2, are thermally insulated with mineral wool to minimize thermal losses, which are expected to be especially relevant at the solar receiver outlet due to the high air temperature.

Temperature and pressure sensors were installed in the main reachable points of the air circuit, according to Fig. 2. Type K thermocouples of class A were used, which were previously calibrated inside a wet stabilized bath well against a reference thermocouple. They showed a total error of ±1.2 °C, for which a total maximum measurement uncertainty can be estimated as ±2.0 °C when installed. Absolute pressure measurement by sensors (Wika A-10) yielded an installed total uncertainty of ±25 mbar after calibration. Volume flow rate is measured at the atmospheric inlet by an airflow sensor (Schmidt 30.015 MPM) with a declared total uncertainty of ±3%. A speed sensor provided by the turbocharger manufacturer was used for turbocharger rotating speed measurement with ±0.05% uncertainty, installed on the compressor housing. A SCADA system was implemented for monitoring the measured values and allowing data logging with a time interval of Δt = 30s, using a Programmable Logic Controller PLC (Unitronic USP-104-B10). Averaging was used when possible to decrease the random error.

3. Numerical models

3.1. Turbocharger

Developed by the automotive industry for enhancing thermal engine performances [20], a turbocharger joins a centrifugal compressor with a turbine, sharing the same shaft and rotating at the rotational speed nT. Newton’s second law of rotating systems, Eq. (1), turns into Eq. (2) under steady-state operation when the mechanical power provided by the turbine Wt drives the compressor, requiring power Wc. A mechanical efficiency ηm accounts for the mechanical losses at the shaft, being Jt the turbocharger polar moment of inertia.

\[
\left( \frac{W_t \eta_m - W_c}{n_T} \right) \frac{1}{\eta_m} = 2\pi \frac{d\omega}{dt}
\]

(1)

where

\[
W_t \eta_m - W_c = 0
\]

(2)

The compressor and turbine power can be calculated using Eqs. (3) and (4), from empirical values of the isentropic total to total efficiencies ηi and ηt, pressure ratios εi = pout/pin, and εt = pint/pout, according to inlet stagnation temperatures Ti and Tt, being γc = c_p/c_v. The outlet temperature Tt and T2 results from Eqs. (5) and (6).

\[
W_c = m_a \epsilon_{C,T} T_i \left( \frac{n_T}{n_{\epsilon_{C,T}}} - 1 \right) \eta_i^{-1}
\]

(3)

\[
W_t = m_a \epsilon_{T,T} T_t \left[ 1 - \sigma^{-1} - \frac{\gamma - 1}{\gamma} \right] \eta_t^{-1}
\]

(4)

\[
T_t = T_i \left[ 1 + \left( \frac{\gamma - 1}{\gamma} \right) - 1 \right] \eta_t^{-1}
\]

(5)

\[
T_t = T_0 \left[ 1 - \eta_i \left( 1 - \sigma^{-1} - \frac{\gamma - 1}{\gamma} \right) \right]
\]

(6)

In the present layout, temperature T3 results from air heating inside the receiver tube as well as across the experimental post-heating unit. Total pressure at the turbine inlet p2 = p3 - Δp is affected by the stagnation pressure drop across the pressurized circuit, Δp = ∑Δp_i, can be estimated using Eq. (7), considering kinetic and frictional components Δp_k, Δp_f, for any portion i of the circuit across the length L_ia, L_iaF. The localized pressure drops introduced by localized connections, elbows, and cross-section variation, are accounted for by K_i.

\[
\Delta p_i = \left( \frac{1}{\rho_{ia}} \right) \left( \frac{1}{\rho_{ia}^2} + \frac{1}{\rho_{ia}^2} \right) \left( \frac{L_i}{D_i} + \frac{K_i}{\rho_{ia}} \right) \Delta \rho_i^{-1}
\]

(7)

The absence of auxiliary compressors results in T1 = T_{in} and p_{in} = p_{h}, with T_{in} = T_{amb} and p_{h} = p_{low} neglecting the inlet nozzle head loss. Otherwise, the auxiliary compressor ratio η_{ac} > 1 leads to p_{in} = η_{ac} p_{in}, whereas T_{in} results from Eq. (8).

\[
T_{in} = T_{in} \left[ 1 + \left( \frac{\gamma_{ac}}{\gamma_{ac} - 1} \right)^{-1} - 1 \right] \eta_{ac}^{-1}
\]

(8)

3.1.1. Compressor

According to [20] among others, the compressor performances are inter and extrapolated from the maps provided by the manufacturer. There is a functional dependence, ηi = ηi(mc, T_in, m_r) and ηt = ηt(mc, T_in, n_r), which are reported in terms of corrected mass flow rate mc,cor and speed mc,cor. According to reference values T_{ref}, p_{ref} for the inlet variables mc, Eqs. (9) and (10), as in Fig. 5(a) for the turbocharger.

(9)

(10)
\[ \dot{m}_{c,\text{cor}} = \dot{m}_{ma,\text{avg}} T_c, \text{in} + \kappa T_{c,\text{ref}} + \kappa \sqrt{p_{c,\text{ref}}}, \text{in} \] (9)

\[ n_{c,\text{cor}} = n_T \frac{T_{c,\text{ref}} + \kappa}{T_{c,\text{in}} + \kappa} \] (10)

Extrapolation of \( \pi_c(\dot{m}_{c,\text{cor}}, n_{c,\text{cor}}) \) is prepared using the methodology proposed in [21]. The non-dimensional flow rate parameter \( \phi_c \) and head parameter \( \psi_c \) are defined through the tip blade speed \( U_c \) and the compressor rotor diameter \( D_c \): \( c_p, T_a, \rho_a, \gamma_c \) are evaluated at the reference temperature \( T_{c,\text{ref}} \): Eqs. (11), (12) and (13).

\[ \phi_c = \frac{\dot{m}_{c,\text{cor}} \rho_c D_c}{U_c^2} \] (11)

\[ \psi_c = \frac{c_p T_{c,\text{ref}} (\gamma - 1) n}{U_c^2} \] (12)

\[ U_c = n_{c,\text{cor}} \pi D_c \] (13)

The fitting function \( \psi_c(\phi_c, n_{c,\text{cor}}) \), proposed by those authors, counts on six parameters \( k_{ij} \) and the inlet Mach number \( Ma \). Here, it has been

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Fig. 5. Compressor map and derived points from the manufacturer data with the model fitted curves superimposed. (a) Compressor map \( \pi_c(\dot{m}_{c,\text{cor}}) \) for \( n_{c,\text{cor}} = \text{const} \) (dots) and dorsal line \( \pi_c(\dot{m}_{c,\text{cor}}) \) for \( \eta_{c,\text{max}} \) (dash). (b) Choke and surge limits, dorsal-speed lines \( \eta_{c,\text{max}}(n_{c,\text{cor}}) \) and \( \dot{m}_{c,\text{max}}(n_{c,\text{cor}}) \). (c) Normalized iso-speed efficiency curves. (d) Iso-speed efficiency curves \( \eta_c(\dot{m}_{c,\text{cor}}), T_{c,\text{ref}} = 29.45 \degree C, p_{c,\text{ref}} = 0.962 \text{bar} \).
replaced with a third-order polynomial form for \( k_i(Ma) \), as proposed by [22], using nine parameters \( k_{ij} \). Eq. (15). The function \( \pi_c = \pi_c(\bar{m}_c, n_{c, cor}) \) results as in Eq. (16). Fig. 5(a) reports \( \pi_c(\bar{m}_{c, cor}, n_{c, cor}) \) on the original compressor map, showing high accuracy.

\[
Ma = U_c / \sqrt{T} RT_c \tag{14}
\]

\[
\psi_c = \frac{k_i + k_2 \phi_c}{k_i - \phi_c} \quad k_i = k_1 + k_2 Ma + k_3 Ma^2 \tag{15}
\]

\[
\bar{\pi}_c = \left[ \frac{1}{(n_{c, cor} \pi D_c)} \psi_c \sqrt{\eta_c(n_{c, cor}, n_{c, cor})} \right] e_c T_c + 1 \tag{16}
\]

The iso-speed efficiency curves \( \eta_c(\bar{m}_{c, cor}, n_{c, cor} = \text{const.}) \) are locally interpolated aiming at high accuracy, keeping simplicity, and a low computational load. Each iso-speed efficiency data set have been normalized through the maximum efficiency value at such speed \( \eta_{c, \text{max}} \), Eq. (17), and expressed as a function of a normalized non-dimensional mass flow rate \( \phi_c \) is defined in Eq. (11) and it is normalized using \( \phi_{c, \text{max}} \) which corresponds to \( n_{c, \text{cor}} \), the abcissa of the here-called dorsal line, depicted in the map joining the \( \eta_{c, \text{max}} \) points.

\[
\eta_{c, \text{norm}} = \frac{\eta_c}{\eta_{c, \text{max}}} \tag{17}
\]

\[
\phi_{c, \text{norm}} = \frac{\phi_c}{\phi_{c, \text{max}}} \tag{18}
\]

Then, each normalized iso-speed data set \( \eta_{c, \text{norm}}(\phi_{c, \text{norm}}) \) is approximated using a second-order polynomial regression, separately for the two regions at both sides of the dorsal line \( \phi_{c, \text{norm}} < 1 \) and \( \phi_{c, \text{norm}} > 1 \) for increasing the accuracy as the behavior at both sides is different, Fig. 5(c), obtaining the functions \( \eta_{c, \text{norm}} = f_{\pi c}(\phi_{c, \text{norm}}) \). An interpolation ratio \( IR \) allows determining \( \eta_{c, \text{norm}} \) for a generic \( \phi_{c, \text{norm}} \) and speed \( n_{c, cor} < n_{c, \text{cor}} < n_{c, \text{cor}} \), Eqs. (19) and (20).

\[
\eta_{c, \text{norm}} = \frac{f_{\pi c}(\phi_{c, \text{norm}})(1 - IR)}{IR} + f_{\pi c}(\phi_{c, \text{norm}}) IR \tag{19}
\]

\[
IR = \frac{n_{c, cor} - n_{c, cor}}{n_{c, cor} - n_{c, cor}} \tag{20}
\]

To ensure continuity between the left and the right region of the dorsal line, \( \eta_{c, \text{norm}} = 1 \) is imposed around \( \phi_{c, \text{norm}} = 1 \). Fig. 5(d) depicts the results. It is worth mentioning that the minimum corrected speed is considered to be \( n_{c, \text{cor}}^{\min} = 110 \text{krpm} \) since not enough points are available on the map for extrapolating an accurate efficiency curve at 90krpm.

3.1.2. Turbine

In general, the manufacturer’s turbine maps report a single curve \( m_{c, cor}(\pi_c) \), as an approximation of the iso-speed mass flow curves. The relation \( m_{c, cor}(\pi_c) \), is modeled according to [20] using Eq. (23), with \( c_e \) and \( k_e \) determined by curve fitting and using as reference \( T_{c, ref} \) and \( p_{c, ref} \), which are given by the manufacturer. Corrected values for mass flow and speed are used, Eqs. (21) and (22).

\[
m_{c, cor} = \frac{P_{c, ref}}{P_{c, cor}} \sqrt{\frac{T_{c, ref} + k_e}{T_{c, cor} + k_e} \pi_c} \tag{21}
\]

\[
m_{c, cor} = \frac{\bar{m}_c}{\bar{m}_{c, cor}} \sqrt{\frac{T_{c, cor} + k_e}{T_{c, ref} + k_e} \pi_c} \tag{22}
\]

The non-dimensional mass flow rate \( \bar{m}_{c, cor} \) data and model results for the turbine. \( T_{c, ref} = 15.7 \text{C}_p, p_{c, ref} = 1.013 \text{bar}, \eta_{c, \text{max}} = 0.62 \).

\[
SR(n_{c, cor}, \pi_c) = U_c / \sqrt{\frac{2 \pi_g \gamma \left( 1 - \pi_c^{\gamma - 1/\gamma} \right)}{\gamma - 1}} \tag{25}
\]

\[
\eta_c(n_{c, cor}, \pi_c) = \eta_{c, \text{max}} \left[ \frac{2SR \eta_{c, \text{opt}}}{SR_{\text{opt}}} \right]^{2} \tag{26}
\]

Fig. 6 depicts the curves of \( m_{c, cor}(\pi_c) \) and \( \eta_c(n_{c, cor}, \pi_c) \) obtained, according to this scheme.

3.2. Diabatic turbocharger

Heat transfer phenomena occurring between compressor, turbine, and the ambient can affect the turbocharger performances and consequently their prediction. Customarily, this is performed using an adiabatic numerical model that ignores them, like the one proposed in the previous sections. Several authors recently claimed their relevance to turbocharger performance. The standard approach uses turbine and compressor experimental maps, provided by the manufacturer through an experimental test bench, which can be “cold” or “hot” according to the temperature of gases during the test. According to (23), maps from the “hot” test include heat fluxes. Their usage leads to accurate predictions only when the operative temperature is the same as the gas reference temperatures used in the test bench. On the other hand, the use of “cold” maps leads to lower accuracy, especially for turbines.
Testing of turbocompressors or turbines alone at near atmospheric conditions, thus "cold", approaches an adiabatic process very well. Dombrovsky [23] stated that a heat fluxes model is needed in both cases if good predictions are demanded and reported several recent works on heat transfer phenomena in turbochargers. Schinnerl et al. [24] recognized that in the low and medium rotating speed regimes, the work transfer is small, making heat transfer more relevant, being this attributed to low mass flow rates and pressure ratios.

Bohn and Heuer [25] and [26] developed a detailed 1D numerical model of heat fluxes besides evaluating them through an experimental analysis. A 1D heat transfer model was also proposed by [23] and [24]. Cormerais et al. [27] experimentally investigated the heat transfer in a turbocharger test bench. They proposed three simple methods for modifying a conventional adiabatic turbocharger model to account for heat fluxes.

Casey and Fesich [28] stated that using the isentropic process as a reference for the ideal work required by the compressor is not justified when the process is not anymore adiabatic. Instead, a polytropic efficiency accounting for the diabatic flows is more appropriate. The low compression ratio variations in the present study do not need such a paradigm, more appropriate for parametric studies.

According to [24] and [26], the isentropic efficiency can still be used combined with the heat transfer model, recognizing that an isobaric heat transfer process occurs before and after the compression/expansion as Fig. 7 indicates, reaching the same starting and ending points. Isobaric heat exchange by the compressor and turbine flow, respectively \( Q_c \) and \( Q_e \), can be split into two components, before and after, respectively \( Q_{c,b} + Q_{c,af} \) and \( Q_{e,b} + Q_{e,af} \). As a consequence, the inlet and outlet total temperatures of diabatic processes \( T_{adi,1} \), \( T_{adi,2} \), \( T_{adi,3} \), and \( T_{adi,4} \) do not coincide anymore with inlet and outlet diabatic total temperatures at compressor and turbine rotor inlet and outlet, respectively, \( T_{1t} \), \( T_{2t} \), \( T_{3t} \), and \( T_{4t} \).

Several heat fluxes can be identified considering a diabatic turbocharger. They slightly vary according to the level of accuracy of the thermal analysis. The simplified study performed in this section considers three main heat fluxes. Heat loss from the compressor flow to ambient \( Q_{c→amb} \) occurs across the compressor case. According to Fig. 8, a heat transfer from the turbine flow to the attached compressor \( Q_{e→c} \) happens through the turbine case, the bearing housing and the shaft, as well as through the compressor case portion consecutive to the bearing house, named the "backplate". Heat transfer to ambient from the turbine flows to the attached compressor \( Q_{e→c} \) happens through the turbine case, the bearing housing and the shaft, as well as through the compressor case portion consecutive to the bearing house, named the "backplate".
flow $Q_{c-amb}$ can include the thermal losses across the turbine case $Q_{c-turb}^{\text{case}}$, across the bearing housing $Q_{c-hub}$, and heat transferred to the lubrication oil $Q_{c-amb}^{\text{oil}}$.

According to [26], $Q_{c-amb}$ can be neglected in front of $Q_{c-turb}$ as a first approximation, since the inlet air flow only crosses a very short passage, the inducer inlet pipe, before compression, and its over-temperature is small. Extending this assumption to the turbine side, $Q_{t-amb}$ can be neglected in front of $Q_{t-amb}$ since the exit flow only goes through a short passage after expansion. These simplifying assumptions lead to $T_{\text{in}}^{\text{ref}} \approx T_1$ and $T_{\text{out}}^{\text{ref}} \approx T_u$.

Accordingly, compressor and turbine heat fluxes can be written as in Eq. (27) and Eq. (28).

$$Q_{c} \approx Q_{c-amb} = \dot{Q}_{c-turb} = Q_{c-turb}^{\text{case}} + Q_{c-hub} + Q_{c-amb}^{\text{oil}}$$

$$Q_{t} \approx Q_{t-amb} = \dot{Q}_{t-amb} = Q_{t-amb}^{\text{case}} + Q_{t-hub} + Q_{t-amb}^{\text{oil}}$$

Estimation of heat fluxes is important to enhance the prediction accuracy of the adiabatic model provided in the previous section, which is based on "cold maps". A semi-empirical quantification of heat fluxes $Q_{c-turb}$, $Q_{c-amb}$ and $Q_{t-amb}$ is discussed in the following section.

From the previous analysis, it derives that Eq. (1) to Eq. (3) keep their validity, whereas Eqs. (4) to (6) must be substituted by Eq. (29) to (31), respectively.

$$W_e = \dot{m}_e c_p e \left[ 1 - \eta_e^{(\gamma_e - 1)/\gamma_e} \right] \eta_e$$

$$= \dot{m}_e c_p e \left( T_n - \frac{\dot{Q}}{\dot{m}_e c_p e} \right) \left[ 1 - \eta_e^{(\gamma_e - 1)/\gamma_e} \right] \eta_e$$

$$T_{\text{in}} = T_{\text{out}}^{\text{ref}} + \frac{\dot{Q}}{\dot{m}_e c_p e} = T_n \left[ 1 + \left( \frac{\gamma_e^{\gamma_e - 1}}{\gamma_e - 1} \right) - 1 \right]^{-1} + \frac{\dot{Q}}{\dot{m}_e c_p e}$$

$$T_{\text{out}} = T_{\text{in}} \left[ 1 - \eta_e \left( \frac{\gamma_e^{\gamma_e - 1}}{\gamma_e - 1} \right) \right]$$

$$= \left( T_n - \frac{\dot{Q}}{\dot{m}_e c_p e} \right) \left[ 1 - \eta_e \left( \frac{\gamma_e^{\gamma_e - 1}}{\gamma_e - 1} \right) \right]$$

4. Results

4.1. Experiments overview

This section gives an overview of two representative experiments, both carried out during the summer season under an optimal clear sky. The T-SAH operation without the experimental post-heating unit is presented first. Then, the T-SAH operation is enhanced with the experimental post-heating unit additional power, showing different working conditions for the setup.

4.1.1. T-SAH Operation along a day

The prototype operated with the auxiliary compressor working in series with the turbocharger compressor at its inlet. The experimental post-heating unit was not activated. The three solar modules were in tracking mode during the considered time interval.

Fig. 9(a) reports temperature at the reference points of the circuit versus True Solar Time TST. While a substantially time-steady profile is obtained for the compressor static inlet temperature $T_1$, following the ambient temperature, the compressor outlet temperature $T_2$ increases along TST following the growth of compression ratio $\pi = p_2/p_1$. Receiver outlet temperature $T_R$ varies along TST according to the solar power received. A peak temperature $T_{TST}$ above 500 °C is reached. A temperature drop from $T_2$ to $T_R$ and from $T_1$ to $T_s$ across the connection pipe lengths $L_{11}$ and $L_{12}$ can be observed. This is also due to the thermal inertia of the pipes $n_1$ and $n_2$, the experimental post-heating unit structure located within the $n_2$ length, and to a lesser extent the thermal insulation and the heat losses across it. After expansion, a peak air temperature at the turbine outlet $T_{turb} = 400$ °C is obtained.

Fig. 9(b) shows the power delivered to the main circuit points. Fig. 9(c) shows the mass flow rate and turbocharger speed. Following the inlet temperature trend, the turbocharger speed reaches a peak of 110 krpm. This coincides with the higher pressure inside the receiver, surpassing 1.4 bar. The mass flow rate varies smoothly, increasing with speed and pressure.

Fig. 9(d) shows the power gained by the air $Q_{\text{in}}$ inside the receiver tube length vs. time, Eq. (32). With the temperature at the turbine outlet $T_{turb}$, the power delivered to the outlet airflow is $Q_{\text{out}}$, Eq. (33). Fig. 9(d) reports the thermal losses $Q_{\text{loss}}$ occurring across the connection tube $n_1$, which were not negligible, as well as the losses $Q_{\text{loss}}$, across the connection $n_2$, Eqs. (34) and (35). Indicating mainly the thermal inertia of the system, both $Q_{\text{in}}$ and $Q_{\text{loss}}$ decrease over time. Because of their thermal capacity, stainless steel connection tubes take between one and two hours to reach their operative temperature, heated by the airflow.

$$\dot{Q}_{\text{in}} = \left( T_{n1} c_{p1} - T_{n2} c_{p2} \right) \dot{m}_{\text{in}}$$

$$\dot{Q}_{\text{out}} = \left( T_{n2} c_{p2} - T_{\text{amb}} c_{p\text{amb}} \right) \dot{m}_{\text{in}}$$

$$\dot{Q}_{t_1} = \left( T_{n2} c_{p2} - T_{n3} c_{p3} \right) \dot{m}_{\text{in}}$$

$$\dot{Q}_{t_2} = \left( T_{turb} c_{p\text{turb}} - T_{\text{amb}} c_{p\text{amb}} \right) \dot{m}_{\text{in}}$$

Some outcomes can be highlighted. This small-scale example of T-SAH demonstrated the viability of direct air heating inside a linear Fresnel collector up to 500 °C with a delivery temperature of 400 °C. The turbocharger increased the air pressure into the T-SAH, recovering compressing power through the turbine. The auxiliary compressor was needed to sustain the turbocharger freewheeling under the operating conditions tested. This is due to two main reasons. On one side, the small turbocharger limits compressor and turbine efficiencies, which in this case have modest values. On the other side, the turbocharger does not operate under optimal conditions. A modest compressing ratio of 1.4 is reached, indicating that the compressor is working in the low region of the map. This is confirmed by the turbocharger speed, which does not overcome 110krpm. The compressor working point is determined by the turbine inlet temperature, which mainly controls the turbine power $W_e$. An increase of $W_e$ would accelerate the turbocharger, shifting its working point toward the map core, where efficiencies are higher and the larger compression ratio would reduce the pressure drops. As a consequence, the mass flow rate would increase, resulting in lower inlet turbine temperature for a given solar power available, thus again slowing down the turbocharger. This behavior highlights the importance of good matching between the turbocharger and solar field. Either, a smaller turbocharger would be more appropriate for the available solar power of the three LFCs, or an LFC field designed for a higher mass flow rate and same outlet temperature would be advisable, keeping the ideas exposed in [15]. Despite the limited performances of the turbocharger, the overpressure imposed by ac decreases for increasing speed, as can be noted by observing $p_{\text{in}}$ in Fig. 9(b).
In this test, the three Fresnel modules were also in tracking mode throughout the experiment. The auxiliary compressor was active. The experimental post-heating unit was activated during the experiment with increasing electrical power. The use of an electrical post-heating unit downstream the solar tube allowed achieving a higher turbine inlet temperature without the risk of receiver tube overheating, hence extending the turbocharger testing range. The effect of the experimental post-heating unit can be seen after the first half-hour on temperature $T_3$, Fig. 10(a). A peak inlet turbine temperature of 570 $^\circ$C was reached. $T_3$ was lower than in the previous case, for a bit lower solar power and a higher mass flow rate. The increased $T_3$ accelerated the turbocharger, up to $\approx 130$ krpm increasing the mass flow rate, as required, Fig. 10(b). This also brought up the pressure in the receiver tube to 1.6 bar, Fig. 10(c). Fig. 10(d) plots the electrical power of the post-heating unit $\dot{Q}_\text{ph}$, as well as the net post-heating power delivered to the airflow $\dot{Q}_\text{n2}$, together with other relevant power values, as for the previous case.

The experimental post-heating unit allows reaching higher inlet turbine temperature, speed, pressure, and mass flow rate so that the compressor working point is shifted toward the map core, where slightly...
higher efficiencies are expected. Also, the increased air pressure should reduce the pressure drop across the circuit, or at least mitigate its growth as a consequence of the increased mass flow rate. The beneficial effect of a higher pressure ratio can be noted by observing the very small mismatch between $p_2$ and $p_3$ in Fig. 10(c), in comparison with Fig. 9(b). The beneficial effect of higher $T_3$ on the turbocharger performance is proven by looking at the auxiliary compressor behavior. A drop of $p_1$ can be observed in Fig. 10(c), which is more relevant than in the previous test. The overpressure given by the auxiliary compressor drops, but it was still necessary to keep the turbocharger freewheeling.
4.2. Diabatic turbocharger analysis

This section performs an in-depth analysis of the turbocharger performances under the two experimental tests reported above. According to the diabatic turbocharger behavior introduced in Section 3, the main heat transfer contributions are estimated as follows, through a semi-empirical analysis, carried out combining the diabatic turbocharger model from “cold” maps with experimental data (temperature and pressure, speed, and mass flow rate).

\[ T_{ad}^{int} \] can be estimated using the modeled compressor efficiency \( \eta_c(T_{int}, m, n_f) \) and the experimental \( \pi_w \) from Eq. (36).

\[ T_{ad}^{int} = \frac{(\pi_w^{(1/-1)} - 1)}{\eta_c(T_{int}, m, n_f)} + T_1 \quad (36) \]

Again, using the turbine efficiency model, the adiabatic turbine inlet temperature \( T_{ad}^{int} \) can be estimated from experimental \( \pi_w \), \( \eta_t(m, T_{ad}^{in}, \pi_w) \) is dependent on \( T_{ad}^{int} \) so that an iterative procedure is needed for solving Eq. (37).

\[ \eta_t(m, T_{ad}^{int}, \pi_w) = \frac{T_{ad}^{in} \pi_w - T_{af} \pi_w}{(1 - \pi_w^{(1/-1)} \pi_w - T_{ad}^{in} \pi_w)} \quad (37) \]

The adiabatic compressor and turbine power result from Eq. (38) and Eq. (39).

\[ \dot{W}_c = \left( T_{ad}^{in} \pi_c - T_1 \pi_c \right) m_a \quad (38) \]

\[ \dot{W}_e = \left( T_{ad}^{in} \pi_e - T_1 \pi_e \right) m_a \quad (39) \]

The diabatic compressor and turbine power come from experimental data, Eq. (40) and Eq. (41).

\[ \dot{W}_c^{adi} = \left( T_2 \pi_c - T_3 \pi_c \right) m_a \quad (40) \]

\[ \dot{W}_e^{adi} = \left( T_2 \pi_e - T_3 \pi_e \right) m_a \quad (41) \]

Estimation of the heat fluxes \( \dot{Q}_{c-amb} \), \( \dot{Q}_{e-amb} \), and \( \dot{Q}_{c-amb} \) can be carried out as follows.

\( \dot{Q}_{c-amb} \) can be estimated by modeling the compressor outlet spiral casing as a tube with an internal diameter \( D_c = (D_{c,in} + D_{c,ou})/2 \) and a length \( L_c = \pi D_{c,ou} \). The internal heat transfer coefficient for convection \( h_{c,ind} \) and external heat transfer coefficient \( h_{c,case} = \dot{q}_{c,mint} + h_{c,rad}^{adi} \) include convection and radiation, neglecting conduction temperature loss across the wall.

Modeling \( Q_{c-amb} \) would be complex due to several heat transfer phenomena occurring on the turbine side, inside the bearing housing, oil lubrication system, and finally inside the compressor case. Instead, it is possible to estimate it from Eq. (42), being \( Q_{e,ad} \equiv Q_e \dot{Q}_{e-amb} \),

\[ \dot{Q}_{e-amb} = \dot{W}_e^{adi} - \dot{W}_e \quad (42) \]

On the turbine side \( Q_{e,ad} \equiv Q_e = Q_{e-amb} + \dot{Q}_{e-amb} \) so that \( \dot{Q}_e \) and \( \dot{Q}_{e-amb} \) can be then obtained from the experimental \( \dot{W}_e^{adi} \) as Eq. (46) indicates, Eq. (43).

\[ \dot{Q}_{e-amb} = \dot{Q}_e - \dot{Q}_{e-amb} = \dot{W}_e^{adi} - \dot{W}_e - \dot{Q}_{e-amb} \quad (43) \]

This methodology was applied to experimental data from the T-SAH operation with post-heating, where the higher turbocharger speeds were reached and the turbocharger operated inside the map with \( n_{c,cor} > n_{c,min} \), allowing evaluation of adiabatic compressor efficiency, as required in Eq. (36).

\( \dot{Q}_{e-amb} \) was obtained for \( n_{c,cor} > n_{c,min} \) and is reported Fig. 11(a). For \( n_{c,cor} > n_{c,min} \), \( \dot{Q}_{e-amb} \) can be approximated with a fitting function \( \dot{Q}_{e-amb}(T_3 - T_2) \), as Fig. 11(a) shows, with the form \( \dot{Q}_{e-amb}(T_3 - T_2) = c_{c,11}(T_3 - T_2)^{m_1} \) resulting in \( c_{c,11} = 1.86 \times 10^{-11} W \cdot C^{-1} \), and \( m_1 = 5.11 \). Using this approximation, \( \dot{Q}_{e-amb} \) can be drawn across the full experimental time interval as Fig. 11(b) reports. \( \dot{Q}_{c-amb} \) and \( \dot{Q}_c = \dot{Q}_{e-amb} - \dot{Q}_{e-amb} \) are also reported. \( \dot{Q}_{c-amb} \) resulted to be much higher than \( \dot{Q}_{c-amb} \), as expected due to the temperature differences \( T_3 - T_2 > T_2 - T_amb \) and the contribution of conduction of the bearing housing.

Fig. 12(a) shows the turbine heat transfer rate. As expected, \( \dot{Q}_{c-amb} \) was the main thermal loss, reaching 2.4kW although an insulating mineral wood layer of 5 cm covered the turbine case. This is due to the high temperature difference \( T_3 - T_amb \) as well as the large heat transfer surface, which includes the turbine inlet, case, and bearing housing. This term also includes the thermal losses to the lubrication oil, which can be relevant due to the continuous oil flow, needed by journal bearings. \( \dot{Q}_c \) is addressed before expansion, as explained, and it translates into a remarkable drop of diabatic turbine inlet temperature, \( T_3^{adi} < T_3 \), which

![Fig. 11](image-url)
causes lower turbocharger performances. Due to the relevance of $\dot{Q}_e$, it is worth extracting an expression for its prediction.

$$\dot{Q}_e = \dot{Q}_{e \rightarrow \text{amb}} + \dot{Q}_{e \rightarrow c}.$$ 

$\dot{Q}_{e \rightarrow \text{amb}}$ can be approximated as $\dot{Q}_{e \rightarrow \text{amb}} = (UA)_{e \text{amb}} (T_3 - T_{\text{amb}})$, with the empirical value $(UA)_{e \text{amb}} = 4.5 \text{ W C}^{-1}$ as Fig. 12(b) shows, giving a good fit for the temperatures of interest.

Fig. 13 (a) compares the modeled adiabatic and diabatic compressor outlet temperatures, as well as the adiabatic and diabatic turbine inlet temperatures obtained. It can be observed that the main effect of the diabatic turbocharger behavior is cooling the inlet air before the work extraction process. Air heating at the compressor outlet does not affect the compressor power $\dot{W}_c$, and it is not relevant due to the modest figure of $\dot{Q}_e$. Fig. 13 (b) plots the compressor and turbine power variation along with the test. The power provided by the turbine $\dot{W}_e$ delivered to the turbocharger shaft is estimated according to Eq (39). It reaches 2.8 kW at the end of the test with maximum $T_3$. The compressor power $\dot{W}_c$ is slightly lower than $\dot{W}_e$ due to the mechanical efficiency of the shaft $\eta_m > 1$. The isentropic turbine power $\dot{W}_{es} = (1 - \pi_c^{-\gamma - 1/\gamma}) T_{3 \text{adi}}^\gamma \rho_{c p} \dot{m}$, which would be provided by an ideal turbine having inlet air temperature $T_{3 \text{adi}}$ is plotted for comparison. The thermal power $\dot{W}_{\text{dia}}$ extracted from the airflow across the diabatic turbocharger reaches up to 5.8 kW and the maximum $T_3$. It confirms that reducing $\dot{Q}_e = \dot{W}_{\text{dia}} - \dot{W}_e$ would be of great
importance for improving the performance of the turbocharger. The isoentropic compressor power $\dot{W}_{cs}$ and diabatic compressor power $\dot{W}_{dia}$ are also reported.

Fig. 14 shows the compressor and turbine adiabatic efficiencies, plotted against the turbocharger speed $n_T$. Fig. 14 also reports the global adiabatic turbine efficiency, including the mechanical efficiency $\eta_{em} = \eta_e \eta_m$, Eq. (44).

The values of $\eta_{em}$ are slightly lower than $\eta_e$ as expected due to $\eta_m < 1$. Then, the mechanical efficiency can be estimated as $\eta_m = \eta_{em} / \eta_e$, as reported in the same plot. $\eta_m$ reached a mean value close to 0.92 at high speeds and decreased for lower speeds down to 0.6. This behavior is reasonable according to the open literature, e.g., [29], among others. As expected, both the compressor and the turbine efficiencies were modest, as typical in small-scale turbochargers, but the low value of $\eta_e$ seems easily improved as the harsh environment of the pulsed flow of piston engines is not present. Ball bearings would increase $\eta_m$.

4.2.1. Mass flow rate

Fig. 15 reports the operative compressor points on the corresponding map for the performed tests, respectively without post-heating (a) and with post-heating (b). In Fig. 15(a) it can be observed that the compressor worked below the minimum speed $n_{c,min} = 110$krpm and $x_{c,exp} < 1.4$. In Fig. 15(b) higher speeds were reached thanks to higher inlet turbine temperatures enabled by the experimental post-heating unit, allowing the compressor to work inside the map core. In both cases, it can be noted that the compressor did not operate near the dorsal line, which corresponds to maximum efficiency for a given $\pi_c$. Instead, the working points appeared on the leftward region, at lower mass flow rates than on the dorsal line. This behavior, not beneficial for the turbocharger mechanical balance, could be induced by the auxiliary compressor characteristic, shown in the following section.

According to the compressor model presented in the previous section, the compressor pressure ratio can be calculated as $\pi_{c,cal} = \pi_c (\dot{m}_c, T_1, n_T)$ using Eq. (16), and experimental inputs $\dot{m}_e, T_1$ and $n_T$. Fig. 16 shows $x_{c,exp}$ for the performed tests against the experimental value $x_{c,exp} = p_2 / p_1$. The comparison reveals the excellent accuracy of the model, being the maximum relative error width of $\pm 5\%$.

Fig. 17 depicts the experimental corrected turbine mass flow rate $\dot{m}_{e,cor} (x_{c,exp})$ obtained from $\dot{m}_e$ and $T_3$, plotted against pressure ratio $x_{e,exp} = p_3 / p_a$. The calculated values using the model $\dot{m}_{e,cor} (x_{c,exp})$ are also reported, showing good agreement with experimental results. Due to the higher inlet temperatures allowed by the experimental post-heating unit, higher $x_{c,exp}$ are reached, Fig. 17(b).
4.3. Auxiliary compressor

This section analyses the auxiliary compressor behavior. Two electrical driven low-pressure compressors (blowers), arranged in parallel, form the single auxiliary compressor unit, Fig. 2. A dedicated inverter feeds them constantly with a maximum frequency $F_Q = 80$ Hz, three-phase AC. They work with equal mass flow rates $\dot{m}_{ac}/2$, globally providing a $\Delta p_{ac}(\dot{m}_{ac}, F_Q)$, according to the manufacturer map of performances and feeding frequency, as Fig. 18(a) reports. As can be seen in Fig. 18(a), the pressure drops across the ac inlet and outlet piping are not negligible. This is due to the high velocity reached in the small section of the connection pipes and localized losses due to tees and elbows. They are estimated as $\Delta p_{0_t}$ at the auxiliary compressor inlet and $\Delta p_{1_t}$ at its outlet, respectively, according to Eq. (7).

Due to $\Delta p_{0_t}$and $\Delta p_{1_t}$, the pressure ratio produced by the blower $\pi_{ac}^*$ results larger than the measured $\pi_{ac} = p_{1_t}/p_{0_t}$, Fig. 18(b).

$$
\pi_{ac}^* = \frac{p_{1_t} + \Delta p_{1_t}}{p_{0_t} - \Delta p_{0_t}} = \frac{p_{0_t} - \Delta p_{0_t} + \Delta p_{ac}(\dot{m}, F_Q)}{p_{0_t} - \Delta p_{0_t}}
$$

(45)
\[ \Delta p_{ac},t = p_{1t} + \Delta p_{t1}(p_0t + \Delta p_{0t}) \]
can be obtained from measured \( p_{1t} \) and \( p_{0t} \) and estimated \( \Delta p_{t1} \) and \( \Delta p_{0t} \), as reported in Fig. 18(a), revealing the working point of the auxiliary compressor on the map. Parasitic pressure drops induced by inlet and outlet piping \( \Delta p_{1t} + \Delta p_{0t} \) are in the same order of magnitude of overpressure required to sustain the turbocharger \( p_{1t} - p_{0t} \). Moreover, \( \Delta p_{1t} + \Delta p_{0t} \) grows with mass flow rate up to half the overpressure imposed by the auxiliary compressors \( \Delta p_{ac},t \).

The efficiency map is not given by the manufacturer. A rough estimation can be performed on experimental data, now considering the real pressure ratio drops \( \pi_{ac}^r \).

\[ \eta_{ac}^* = \frac{\dot{W}_{ac}}{\dot{W}_{ac}} = \left( \frac{\pi_{ac}^r}{\pi_{ac}} - 1 \right) \frac{T_0^c}{T_{1t}^c} - \frac{T_0^c}{T_{0t}^c} \]  
\( \Delta p_{ac},t \)

Fig. 18(b) depicts results from experimental data, an average value of \( \eta_{ac}^* \approx 0.45 \) was found, in accordance with similar side-channel blowers models.

The auxiliary compressors offer the main effect of increasing \( p_{1t} \), the inlet pressure at the turbocharger, providing the auxiliary mechanical power required by the turbocharger to keep freewheeling. As discussed in the following section, the aid of auxiliary compressor would be avoided using turbochargers with higher efficiency, both in the turbine and compressor side, minimizing thermal losses across them, and increasing \( \eta_{m} \).

In the present setup, the overpressure \( p_{1t} - p_{0t} \) is needed to sustain the turbocharger freewheeling. Moreover, a decreasing value pressure \( p_1 \) is detected as shown in Fig. 10(b), resulting in a decreasing \( \pi_{ac} \) in Fig. 18(b). The corresponding isoentropic mechanical power required \( W_{ac} \) results from Eq. (47).
Besides electrical power consumption, the drawback of using an auxiliary compressor is the increased temperature \( T_3 \), which reduces the turbocharger compressor performance. The non-negligible pressure drops and the low efficiency increase this adverse effect, since \( \pi_{ac} > \pi_{ad} \) leads to a \( T_3 \) increase, as can be seen in Fig. 9(a) and Fig. 10(a), where it ranges between 60 °C and 80 °C.

As Fig. 20 reports, \( W_{ac} \) represents around 18% of thermal power delivered to the air, \( Q_a \), at the beginning of the test, decreasing down to 8%, although the \( W_{ac} \) is much lower in relative terms. Other main power contributions are shown in relative terms over \( Q_a \). The turbine power \( W_e \) is around 10 to 15% of \( Q_a \), while the overall thermal power that is taken from the heated airflow \( W_{dia} \) is twice, up to 30%. In fact, as discussed above, thermal losses from the turbine housing, bearing housing, and through the lubrication oil to the ambient \( Q_{aux,amb} \) are relevant, up to 15% of \( Q_a \). Fig. 20 also reports a thermal efficiency parameter, estimated as \( \eta_{th} = Q_a/(Q_a + Q_{ph} + W_{ac}) \). It does not take into account the thermal efficiency of the solar receiver tubes neither the optical efficiency of the solar collector but only accounts for the thermal losses across insulated piping and the turbocharger. It can be seen that \( \eta_{th} \) grows during the test, while the transient heating effect due to their thermal inertia disappears, reaching \( \eta_{th} \approx 0.84 \). \( Q_{aux,amb} \) is the main cause of thermal efficiency drop, whose reduction is of great importance for the optimal scaling up of the system T-SAH herewith proposed as optimal.

### 4.4. Turbocharger freewheeling issues

According to [15] and [16], minimizing the energy consumed by the auxiliary compressor is one of the main goals for T-SAH. In fact, the correct operating conditions would be achieved when the turbine provides at least enough power for driving the compressor, allowing turbocharger freewheeling without the auxiliary compressor aid, and eventually leaving surplus pressure for head losses down to the consumption point. The auxiliary compressor would be needed only for some transients, such as starting, failures, or temporary drops of solar power.

Using the developed numerical model, it is possible to estimate under which conditions the auxiliary compressor aid can be avoided. The mechanical balance on the turbocharger shaft can be expressed as in Eq. (48), using Eq. (2), (3), and (29), Eqs. (48) and (49) allow the calculation of the inlet turbine (adiabatic) temperature required for providing enough turbine power to drive the compressor.

\[
\eta_{th} = \frac{\eta_{th}^* - \left[1 - \frac{\pi_{ad}^{(2/\gamma - 1)} - 1}{\eta_{th}^* \eta_{a} - \pi_{ad}^{(2/\gamma - 1)}\eta_{a} - 1} \right] \eta_{ph} T_{th} \left(\pi_{ad}^{(2/\gamma - 1)} - 1\right)\eta_{a}^{-1}}{W_{ac} = 0}
\]

\[
T_{th}^* = \frac{T_{th}}{\eta_{th}^* \eta_{a}} \left[1 - \frac{\pi_{ad}^{(2/\gamma - 1)} - 1}{\eta_{a} - \pi_{ad}^{(2/\gamma - 1)}\eta_{a} - 1} \right] \eta_{ph} T_{th} \left(\pi_{ad}^{(2/\gamma - 1)} - 1\right)\eta_{a}^{-1}
\]

The turbine expansion ratio \( \xi = \rho_a/\rho_{amb} = (\rho_a - \Delta\rho_a)/\rho_{amb} \) is affected by the stagnation pressure drop. Pressure drops \( \Delta p_a \), as well as pressure ratios \( \pi_a, \pi_{ad} \) and efficiencies \( \eta_a, \eta_{th} \) depend on the mass flow rate \( m_a \). For each considered turbocharger speed, the mass flow rate results from matching turbine and compressor maps, Eq. (16) and Eq. (23). \( T_{th}^* \) is the independent variable iteratively adjusted to obtain \( W_{ac} = 0 \). Considering a diabatic turbocharger, the inlet temperature \( T_{th} \) would be larger due to the cooling effect of turbine thermal losses \( Q_a \).

Considering the setup experienced, the turbine temperature required for freewheeling without the auxiliary compressor aid, at any turbo-
charger speed in the range of compressor map, was estimated according to Eq. (48). For this purpose, the turbocharger model developed and tuned above has been used. A value of mechanical efficiency \( \eta_m = 0.92 \) was used. The main thermal losses of the diabatic turbocharger at the turbine inlet were taken into account. The stagnation pressure drops across the air circuit were estimated according to Eq. (7). Solar power gain across the solar collector was assumed \( \dot{Q}_{u} = 17 \text{ kW} \) heating the airflow up to \( T_{3r} \), whereas an adjustable post-heating power \( \dot{Q}_{ph} = m(T_{adi3}c_p - T_{adi3}c_p) \) heats the air up to \( T_{3} \). Inlet diabatic turbine temperature is \( T_{3l} = \frac{T_{adi3}}{\eta_m} \), where \( Q_e = Q_{e-amb} + Q_{e-c} = \epsilon_{e,1}(T_3 - T_2)^{ec_1} + (UA)_{amb}(T_3 - T_{amb}) \).

![Graphs showing various parameters](image)

Fig. 21. Freewheeling conditions. (a) Temperatures. (b) Pressure ratios and mass flow rates. (c) Thermal powers and losses. (d) Efficiencies and total pressure drop.

As a general trend,
the equations show that $T_3^{ad}$ is mainly affected by turbocharger overall efficiency $\eta_T = \eta_{\pi} \eta_{\text{m}},$ which varies according to turbocharger speed. When the turbocharger works in the core of the compressor map $130\text{krpm} < n_T < 170\text{krpm}, \eta_{\pi}$ and hence $\eta_T$ reach their maximum, $\eta_T \cong 0.69$ and $\eta_T \cong 0.39,$ respectively, allowing lower inlet turbine temperatures, according to Eq. (49), as reported in Fig. 21(d). Besides, the stagnation pressure drop across the pressurized circuit $\Delta p_{\text{p}}$ has an impact on $T_3^{ad}.$ As speed increases the operating mass flow rate increases, Fig. 21(b), according to the compressor and turbine flow maps. Higher mass flow rate leads to higher stagnation pressure drops $\Delta p_{\text{s}},$ reducing the turbine pressure ratio $x_{\pi}.$ This effect is appreciable, although the increased compression ratio in Fig. 21(b) mitigates the growth of $\Delta p_{\text{p}},$ by reducing air velocity inside the pressurized circuit. These combined effects give that the lowest inlet turbine (adiabatic) temperature of free-wheeling would be $T_3^{ad} = 620\degree C,$ for $n_T = 130\text{krpm},$ with $m_a = 0.035\text{kg}\text{s}^{-1},$ and $x_{\pi} = 1.6.$

The diabatic inlet turbine temperature is above $T_3^{ad}$ due to heat losses $Q_{\text{w,turbo}}$ and $Q_{\text{s,-turbo}}.$ Both of these losses are not negligible and increase with $T_3^{ad},$ as in Fig. 21(c), being in the same order of magnitude of $W_c.$ Fig. 21 (a) gives the outlet receiver temperature resulting from the heating effect of $Q_c.$ The outlet turbine temperature goes up in the range of 550 to 600 $\degree C.$

As Fig. 21 clarifies, turbocharger freewheeling without the auxiliary compressor aid requires inlet adiabatic turbine temperature $T_3^{ad} > 600\degree C$ which wasn’t reached during the experiments carried out. Although turbocharger freewheeling is foreseen to be achieved with null auxiliary energy consumption in a well-designed and/or large installation, reaching it in a small size installation, as the prototype here studied, is challenging, due to low turbocharger efficiency, especially notable the one of the turbine, which is lower than the one of the compressor. Moreover, the diabatic behavior of the turbocharger, especially the thermal losses occurring from the turbine to ambient and to the compressor, makes it difficult to achieve the required turbine power production.

Nevertheless, a decrease in the overpressure imposed by the auxiliary compressor can be noticed from the reported data, either in the case of the collector heating alone or in the case of the post-heating experimental operation. The progressive drop of the auxiliary compressor overpressure $p_{\text{H}}$ as well as the auxiliary isentropic specific power $w_{\text{aux}}$ needed for air pumping underpins what would occur in a more performing installation. After a starting transient, during which the auxiliary compressor is used for accelerating the turbocharger, the inlet compressor pressure would progressively decrease down to ambient pressure, as soon as the inlet turbine temperature reaches the free-wheeling theoretical temperature, Eq. (49). At that point, the auxiliary compressor can be either kept working with negligible power consumption, ready for action, or either bypassed and switched off.

4.5. Advantages of scaling up

The analysis carried out on the experimental setup combining experimental and numerical results allows pointing out several considerations dealing with the foreseeable scaling up the turbo-assisted solar air heater concept T-SAH for an industrial application.

The turbocharger efficiency is of great importance for the viability of the system aiming at the elimination of auxiliary energy consumption for air pumping. In fact, the temperature required at the turbine inlet for enabling turbocharger freewheeling without auxiliary compressor aid is higher as lower is the turbocharger efficiency $\eta_T = \eta_{\pi} \eta_{\text{m}},$ according to Eq. (48). Larger installation than the presented setup would offer a better match with larger size turbochargers which have higher efficiency, up to $\eta_T \cong 0.6,$ according to datasheets of the models available on the market, for Reynolds number and relative roughness considerations in turbomachines scale-up, e.g. [31,32] or [33]. In that case, turbocharger autonomous freewheeling could be achieved at a lower $T_3^{ad},$ with the obvious no need for a post-heating unit, according to the original layout proposed by [15], as demonstrated in the detailed simulation presented in [16]. Using a large size and high-performing turbocharger, parallel collector rows can be considered for scale-up. Attention must be paid to collector length since this affects the pressure drop inside the pressurized circuit which plays against the turbocharger mechanical balance. Pressure losses should be kept to a minimum.

It is worth mentioning that the temperature drop across the expansion process is modest, 100 to 150 $\degree C,$ allowing a temperature of delivery to the user in the range of 300 to 400 $\degree C$ when the inlet turbine does not surpass 500 to 550 $\degree C,$ which is achievable inside the receiver, considering current thermal limits.

The analysis of undesired heat losses to ambient suggests paying attention to a good turbine thermal insulation as well as to a proper lubrication system that minimizes the work dissipation, e.g. roller or air bearings.

Specific speed considerations call for a turbocharger trimmed or optimized for moderate $x_{\pi}$ and $x_{\pi},$ typically lower than current values in automotive turbocharging. The necessity of an auxiliary compressor and its deleterious effect suggest its avoidance and aim at an electrically boosted turbocharger, now emerging in the industry.

5. Conclusions

In the present study, an implementation of the innovative concept of a solar direct air heater assisted by a turbocharger T-SAH is experimentally investigated at a laboratory small-scale using commercial components.

Emphasis is put on the turbocharger characterization as this component plays a crucial role in eliminating the blowing electrical power and even expand to a higher than atmospheric pressure, leaving blowing power for the user’s needs.

A low-capacity automotive turbocharger has been modeled and experimentally characterized under non-standard operating conditions using the advanced diabatic approach.

According to recent researches, a detailed model of the turbocharger is developed integrating the conventional approach based on the compressor and turbine cold maps provided by the manufacturer with a semi-empirical characterization of heat transfer phenomena affecting its performance, the diabatic turbocharger. The model has been verified and tuned using experimental data directly obtained in the facility constructed, thus offering ways of fast and low-cost commissioning of similar facilities.

The small-scale facility did not allow to reach optimum operating conditions in the implementation here considered, due to low turbocharger efficiency, typical of small size turbochargers. Nevertheless, the analysis carried out indicates the relevant features of the system when scaled up to typical sizes.

The results corroborate the practical possibility of an industrial-scale application of the T-SAH concept. Key information has been generated so that demonstration plants are now possible.

Medium-scale industrial applications of T-SAH offer great potential for hot air production, reducing cost, weight, bulk, complexity, and environmental impact over more conventional approaches.
Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgments

This research was supported by the Industrial Ph.D. program of Comunidad de Madrid, Spain (BOCM Reference IND2017/AMB7769).

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