

DRAFT: Energy Conversion Mechanisms in Turbomachines for High-Temperature Endothermic Reactions: Redefining Energy Quality

Dylan Rubini¹

Oxford Thermofluids Institute,
University of Oxford,
Department of Engineering Science.,
Oxford OX2 0ES, UK
email: dylan.rubini@eng.ox.ac.uk

Budimir Rosic

Oxford Thermofluids Institute,
University of Oxford,
Department of Engineering Science,
Oxford OX2 0ES, UK
email: budimir.rosic@eng.ox.ac.uk

Liping Xu

Whittle Laboratory,
University of Cambridge,
Department of Engineering,
Cambridge CB3 0DY, UK
email: lpx1@cam.ac.uk

Through the growth of a new field of applications for turbomachines, an entirely new aerothermochemical design space has been revealed. The most complex of these new applications is the decarbonization of hard-to-abate high-temperature endothermic chemical reaction processes. This paper explains how the novel turbo-reactor, developed for these industries, has the potential to be “chemically reaction tuned” for the specific reaction dynamics. By leveraging aerodynamic losses, the energy-supply rate distribution into the chemical reaction can be designed to match an ideal reaction-efficient temperature profile while simultaneously promoting flowfield uniformity. This is not possible in conventional furnaces. Exploiting losses in this way is an inherently new approach to turbomachinery design. Through the lens of chemical exergy, this paper clarifies why degrading the energy quality of the electricity supply is a logical design philosophy. Fundamentally, the mechanical energy imparted by the rotating blade row is converted into internal energy to drive the reaction primarily through viscous mechanisms. Therefore, to optimize the aerodynamic design of the bladed flow path to match an ideal energy transfer and transformation profile, the physics and breakdown of the energy conversion mechanisms must be understood. This study achieves this by exploiting URANS and LES to numerically investigate the breakdown of these mechanisms over a range of flow regimes.

Keywords: Turbomachinery, LES, exergy, loss mechanisms, energy conversion devices, high-temperature industrial processes, reacting flow, pyrolysis, turbo-reactor

1 Introduction

1.1 Historical Perspective, Motivation and Background.

The fundamental principles of turbomachinery design have been shaped to minimize the degradation of energy quality, *i.e.*, the exergy destruction or lost work potential, during energy conversion processes in power generation, aviation and industrial processes. The focus on reducing energy quality degradation (and increasing fuel efficiency) intensified following the 1970s oil crisis, reaffirming that the finite fossil fuel resource was rapidly depleting, and encouraging industries to strive towards extracting the maximum utilizable work, that is, the exergy, from nonrenewable energy resources. Today, the drive to minimize aerodynamic losses for efficiency gains is ubiquitous in turbomachinery.

In the context of conventional compressors (*i.e.*, energy-imparting machines), the bladed flow path is designed to maintain the absorbed shaft power in the form of pressure or kinetic energy to preserve energy quality. However, now with the emergence of new applications for turbomachines—decarbonizing high-temperature endothermic reaction processes with the new turbo-reactor [1–3]—this is no longer the case. Specifically, the goal of the turbo-reactor seen in Fig. 1(a) is to convert energy from the electric-motor-powered shaft into internal energy of the fluid (not pressure). Since there is no global pressure rise, this can be achieved using ultrahigh-loading rotating blades to impart mechanical energy to the fluid before it is dissipated (converted) into internal energy [4, 5]. The working fluid undergoes a chemical reaction along most of the bladed flow path, which presents a new set of requirements for the design. That is, the imparted mechan-

ical energy must be dissipated at a rapid rate determined by the chemical reaction. This means that enhanced entropy generation is unavoidable. This new turbomachinery design principle of leveraging entropy generation for ultrafast energy transformation along the bladed path is fundamentally different from conventional wisdom.

1.2 Objectives of this Paper. Following a recap of the basic requirements and key features of the turbo-reactor concept [1–5], the paper is organized into four distinct parts summarized below.

- (1) The reaction requirements are outlined along with how the turbo-reactor is designed to achieve them.
- (2) The consequences of converting electricity to heat and thus degrading energy quality are clarified. By introducing chemical exergy as a new vector of energy quality for turbomachines, it will be demonstrated that this process is sensible from a thermodynamic perspective.
- (3) This paper demonstrates how the turbo-reactor can be designed as a “chemically reaction tuned”—or for brevity, a “chemically tuned”—device (and is not a “bad compressor” or a “mixer”). It is explained that aerodynamic losses can be exploited to design a reaction-efficient temperature profile by controlling the dissipation profile. This unlocks a new dimension within the design space of turbomachines.
- (4) To design this reaction-efficient temperature profile, an improved understanding of the energy conversion mechanisms is required. Therefore, the distribution of energy conversion

¹Corresponding Author.

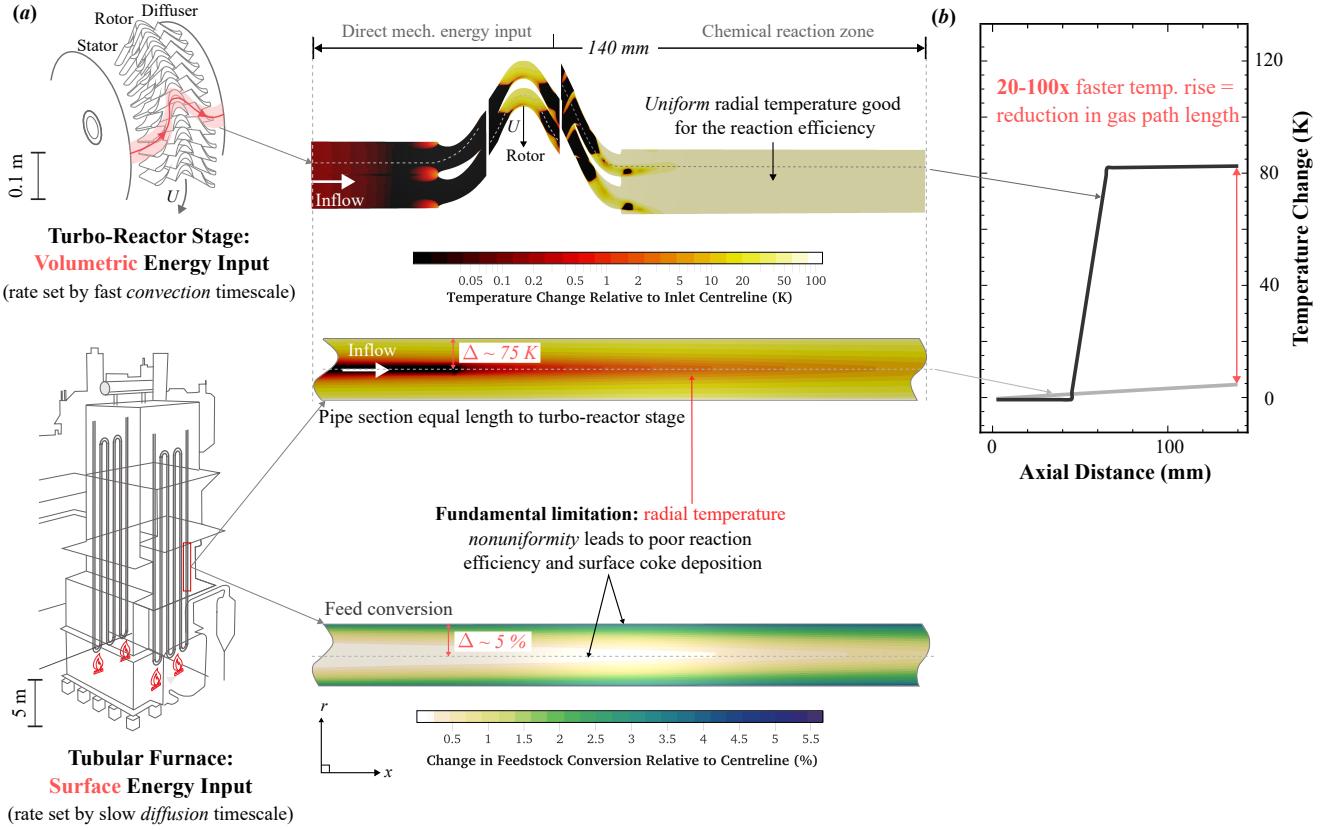


Fig. 1 Comparison between a turbo-reactor stage and a typical steam cracking furnace pipe, showing: (a) static temperature change on a logarithmic scale and change in ethane feedstock conversion and (b) the averaged axial temperature change

mechanisms is numerically investigated over a range of design conditions. An understanding of this process motivates further aerodynamic design optimization to tune the energy conversion process for increased reaction efficiency.

The focus of this paper is on reacting flow applications. For context, the example endothermic reaction application of steam cracking of hydrocarbon feedstocks [1, 2] is used. However, Rubini *et al.* [5] demonstrated that the turbo-reactor can also be used to decarbonize a variety of other endothermic reaction processes.

2 Decarbonizing Endothermic Reactions: The Turbo-Reactor Concept

The renewably-powered turbo-reactor replaces fossil-fuel-fired tubular furnaces (see Fig. 1). In these furnaces, the rate of energy transfer is limited by the thermal diffusion timescale of surface heat exchange. Moving to direct volumetric energy input within the turbo-reactor (see Fig. 1), the energy transfer rate is instead driven by the rotor convection timescale. Since this timescale is minimized at the high supersonic flow speeds exploited in the turbo-reactor, orders of magnitude higher power densities can be achieved relative to conventional furnaces (see Fig. 1(b)).

2.1 Reaction Requirements. For turbo-reactor-powered endothermic chemical reaction processes, value is created by chemically transforming low-value feedstock (*e.g.*, ethane) into higher-value products (*e.g.*, ethylene). Alongside power input and power density targets, the efficiency of this chemical transformation—that is, the fraction of feedstock that is converted into high-value product species—is the most important figure of merit and displaces isentropic efficiency as the driving design metric for this new class of machines. Achieving a highly efficient chemical reaction requires two design features, both of which are possible by replacing furnaces with the turbo-reactor.

(I) The reaction should progress uniformly in the pitchwise and spanwise directions. The progress of the reaction is most significantly impacted by the temperature field. In the pipes of conventional furnaces, Fig. 1(a) illustrates that the temperature distribution is highly nonuniform due to heat transfer through the walls, resulting in nonuniform reaction progress (*i.e.*, feed conversion) as shown in Fig. 1(a). This leads to poor reaction efficiency and solid carbon (*i.e.*, coke) forming directly on the metal surface. Fundamentally, this challenge of nonuniformity cannot be avoided even with novel electric or hydrogen furnaces [6], which are still surface heat exchangers. In contrast, in the turbo-reactor, the volumetric energy input mechanism combined with turbulent mixing between stages facilitates a significantly more uniform temperature distribution (see Fig. 1(a)). This can promote higher reaction efficiency.

(II) The reaction efficiency (*i.e.*, yield) increases with short reaction timescales at higher static temperature levels, as shown in Fig. 2. Bypassing metallurgical limitations due to the hot walls of conventional surface heat exchanges [1, 2, 4] allows the turbo-reactor to achieve higher temperatures over a shorter reaction timescale. This improvement in reaction efficiency is a key advantage of the turbo-reactor that is not possible with surface-heat-exchange-based methods to decarbonize high-temperature chemical processes (*e.g.*, hydrogen or electric furnaces) [6].

We will see that these features of the turbo-reactor design are enabled by enhancing aerodynamic losses and viscous dissipation.

The next section summarizes the elemental stage design philosophy and approach to configuring the multistage environment to achieve the reaction requirements presented in this section.

2.2 Working Principles of the Turbo-Reactor.

2.2.1 Energy Transfer. Short reaction timescale requirements dictate that an ultrahigh loading impulse rotor is employed. Since there is no requirement for a global pressure gain, it is possible

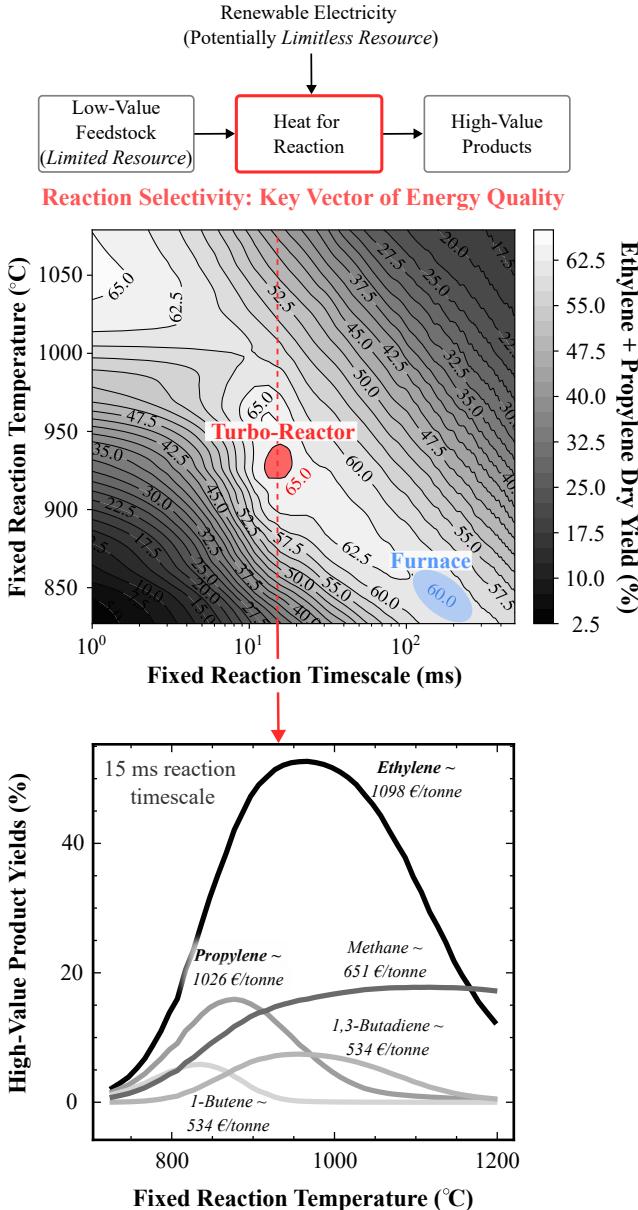


Fig. 2 Idealized dry yields of selected products as a function of fixed reaction temperature and timescale

to develop supersonic rotor blades with a work coefficient of up to $\psi = \frac{\Delta h_0}{U^2} \approx 8.0$, depending on the fluid properties. This is 20 times that of an axial compressor. Without a global pressure rise, efficiency and stability restrictions [7] that limit the work coefficient in compressors are alleviated, enabling this loading level.

2.2.2 Energy Conversion. To supply an enthalpy dh to the chemical reaction at a constant pressure, the Gibbs equation (Eq. 1) below implies that entropy ds must be produced to satisfy the constant global static pressure ratio ($dp \approx 0$) boundary conditions.

$$dh = \frac{1}{\rho} dp + T ds \quad (1)$$

Since the designer is no longer concerned with efficiency, aerodynamic losses (and hence entropy production and viscous dissipation) can be leveraged to control the energy conversion profile. It will be demonstrated in Sec. 3.1 that the conversion process can be tuned to design reaction-efficient temperature profile.

2.2.3 Chemically Tuning the Multistage Arrangement. Figure 3 shows that each stage is separated by an interstage vaneless mixing space where the mechanical energy absorbed through the rotor is converted into internal energy. This energy conversion process must be distributed over a series of stages to counterbalance the drop in temperature due to the endothermic chemical reaction, as shown in Fig. 4. In Fig. 2 it was suggested that the local residence time favored by the reaction varies with the temperature level, which is different for each stage. Therefore, the local residence time can be controlled by varying the length of the vaneless space between stages (L). This becomes a new tunable control parameter (see Fig. 3), which can be customized stage by stage from length 0 to NC_x to design a reaction-efficient temperature profile (see Fig. 4). The design of the temperature profile for efficient chemistry is explained in more detail in Sec. 3.1. Furthermore, this vaneless mixing space is also critical to allow thermochemical nonuniformities to mix-out.

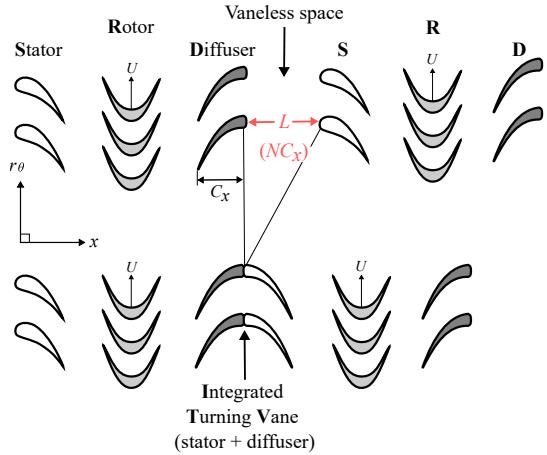


Fig. 3 Variable interstage vaneless space length from 0 diffuser chords (C_x), i.e., an integrated turning vane, up to NC_x

For preheating the gas mixture before reaction activation, there are no reaction timescale restrictions, so a “compact stage” design can be used. This is achieved by combining the stator and diffuser into an integrated turning vane (ITV), as shown in Fig. 3.

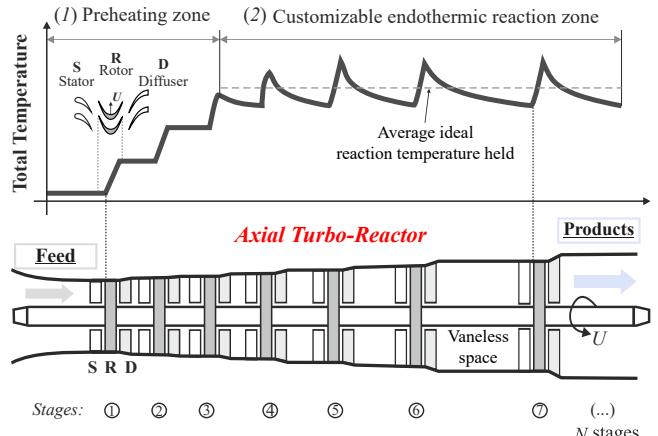


Fig. 4 Schematic of the elemental stage and multistage environment for the turbo-reactor, illustrating a typical total temperature profile for an endothermic chemical process

2.3 Energy Quality (Exergy). It is evident that the chemical processes powered by the turbo-reactor rely on irreversibilities to rapidly dissipate mechanical energy within short reaction

timescales (see Fig. 2) while promoting uniformity. Therefore, it is important to assess the thermodynamic implications of this degradation of energy quality (exergy).

Prior to this, a new vector of energy quality must be introduced for this new class of turbomachines. Conventionally, turbomachinery design practice concentrates only on *physical* exergy. However, in endothermic reaction applications, it supplies energy to a chemical reaction along the bladed flow path. This means that *chemical* exergy, defined in Appendix A, must be considered in addition to physical exergy as a new vector of energy quality. This is an entirely new design consideration for turbomachines.

Equipped with chemical exergy, as well as the price of energy & and the feed/products, the impact of exploiting viscous dissipation can be understood using Fig. 5. It suggests that since the electricity input has significantly lower cost and exergy content compared to the feedstock, and is also semi-limitless (*i.e.*, renewable energy), it is reasonable to sacrifice its energy quality by developing irreversibilities to enhance reaction efficiency. After all, a higher reaction efficiency means more species in the effluent gas stream that have a high value. Therefore, the reaction efficiency can be viewed as a new vector of energy quality.

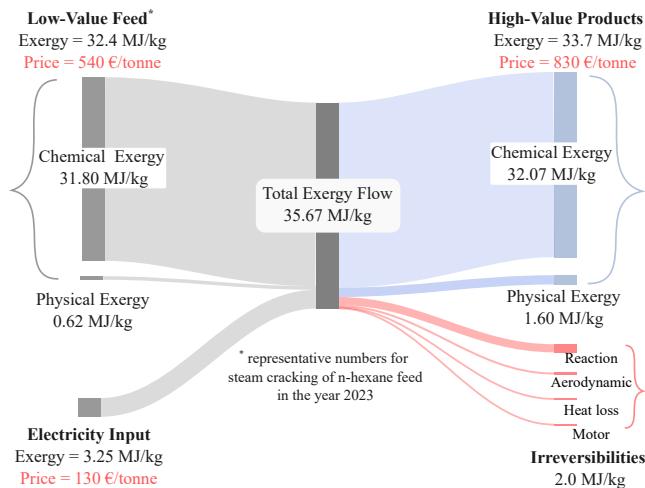


Fig. 5 Sankey diagram showing the an exergy balance for a representative steam cracker with an n-hexane feed in 2023

3 Leveraging Aerodynamic Losses for a Controllable Energy Conversion Profile

3.1 Chemically-Tuned Turbomachinery: Design of a Reaction-Efficient Temperature Profile. In Sec. 2 it was highlighted that at a global coarse-grained level, the turbo-reactor is designed to operate at a higher temperature *level* over shorter timescales to improve reaction efficiency. We saw how to control the temperature at the coarse-grained stage-level (see Sec. 2.2.3) by customizing the interstage vaneless space distance.

At a fine-grained level, there is an optimum ideal temperature profile that benefits reaction performance. This section explains how the local energy conversion process, and thus the profile of energy absorbed by the reaction, could be tuned to “design” an ideal reaction-efficient temperature profile. We will explore the potential benefits of tailoring the energy conversion profile at a fine-grained level, opening new possibilities for aerothermal design optimization. The ability to perform temperature profiling is an important benefit of distributing energy addition across a series of stages.

The following discussion will be structured following the three boxes shown in Fig. 6.

3.1.1 Box ①: Objectives. In an idealized scenario, the temperature level is raised and maintained (see Fig. 6) at an ideal reaction temperature set by the given residence time (see Fig. 2). To better understand how to design for this reaction temperature profile, it is useful to consider the temperature transport equation for the time-averaged base flow [8]. For an ideal gas in simplified form, this is defined as follows:

$$\bar{\rho} c_p \bar{u}_j \frac{\partial \bar{T}}{\partial x_j} \approx \underbrace{\lambda_{eff} \frac{\partial^2 \bar{T}}{\partial T^2}}_{\text{diffusion}} + \underbrace{\bar{u}_i \frac{\partial \bar{p}}{\partial x_i}}_{\text{pressure work}} + \underbrace{(\bar{\tau}_{ij} + \tau_{ij}^R) \frac{\partial \bar{u}_i}{\partial x_j}}_{\text{viscous work}} - \underbrace{\dot{\omega}_T}_{\text{heat sink}} \quad (2)$$

where $\bar{\tau}_{ij}$ is the stress tensor of the mean flow, τ_{ij}^R is the Reynolds stress tensor, \bar{u}_i is the velocity vector, and μ is the molecular dynamic viscosity. The terms in red convert mean flow kinetic energy into enthalpy through pressure work and molecular & turbulent viscous shear work (*i.e.*, viscous dissipation). The term in blue $\dot{\omega}_T$ is the endothermic heat sink rate and is defined as

$$\dot{\omega}_T \approx \sum_{k=1}^{N_{spe.}} \Delta h_{f,k}^0 \cdot \dot{\omega}_k \quad (3)$$

where $\dot{\omega}_k$ are the net species production rates and $\Delta h_{f,k}^0$ the standard formation enthalpies.

Crucially, from Eq. 2 it can be seen that to maintain a constant idealized reaction temperature level ($\frac{DT}{Dt} = \frac{\partial \bar{T}}{\partial x_j} = 0$) subject to temperature sinks due to the endothermic reaction, the combination of pressure work and viscous dissipation must be designed to approximately balance the endothermic sink term $\dot{\omega}_T$ at each streamwise location, as illustrated in Fig. 6.

3.1.2 Box ②: Energy Conversion Mechanisms. The aerodynamic designer is trying to shape the turbo-reactor flow path to achieve an energy supply profile to the reaction such that an ideal reaction-efficient temperature profile is matched. Box ② in Fig. 6 shows the energy conversion profile required to maintain some idealized temperature profile shown in Box ①. Through a series of discrete stages, the turbo-reactor attempts to approximate it.

There are reversible (isentropic) and irreversible (entropic) mechanisms responsible for this energy conversion profile. The first term in red in Eq. 2 is the reversible pressure work $\bar{u}_i \frac{\partial \bar{p}}{\partial x_i}$. This is a consequence of changes in the flow area and the reversible pressure rise component of the shock action.

The remaining energy conversion is provided by irreversible (entropic) energy conversion mechanisms through viscous shear work. This is accounted for by $(\bar{\tau}_{ij} + \tau_{ij}^R) \frac{\partial \bar{u}_i}{\partial x_j}$ in Eq. 2. Some of the primary physical mechanisms are as follows are highlighted at the bottom of Box ② in Fig. 6.

3.1.3 Box ③: Reaction Performance. Controlling the balance between the endothermic sink and the dissipation profile at each streamwise location to better match an ideal reaction-efficient temperature profile can lead to improved reaction efficiency, as illustrated in box ③ of Fig. 6. To understand the importance of improving the reaction efficiency (*i.e.*, the yields), it is highlighted that for a typical hydrocarbon cracking plant (with capacity = 1 million tonnes per year), a 10% improvement in reaction efficiency can increase annual profits by €90 – €110 million (as of 2024). This underscores the importance of tailoring the aerothermal design for improved yields.

It should be noted that this approach of striving for a *constant* ideal reaction temperature is not necessarily the current design strategy or optimal for a general chemical reaction. However, it is a useful way to think about design goals: balancing viscous dissipation and reaction heat sink to produce a reaction-efficient

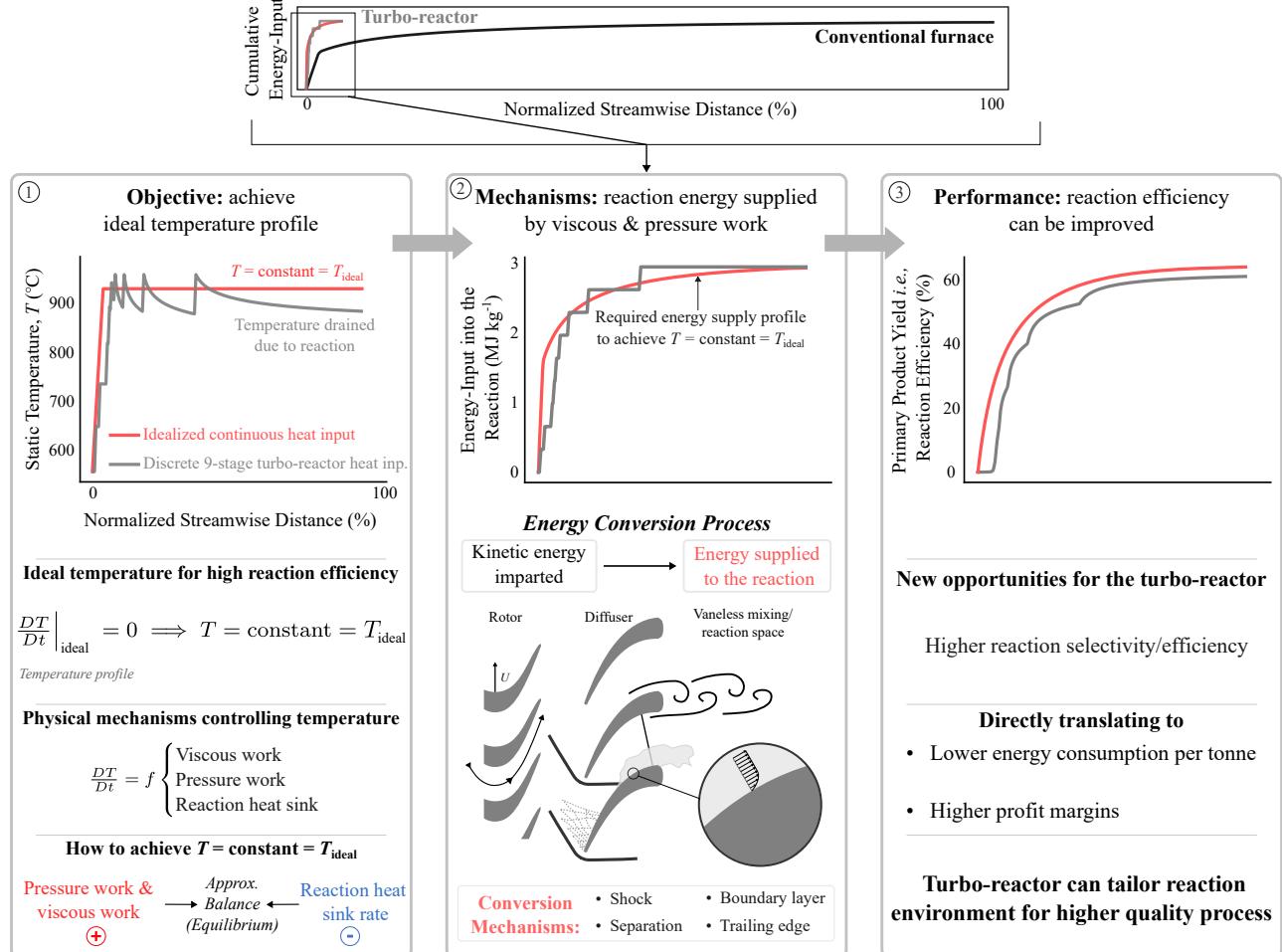


Fig. 6 Designing a reaction efficient temperature profile along the turbo-reactor bladed flow path

temperature profile (see Fig. 6). For different feedstocks or reactions, the ideal temperature level may vary and/or may not be constant, leading to a nonequilibrium balance.

3.2 Theoretical Limit on the Rate of Energy Conversion. To understand the potential range of temperature profile control, it is useful to determine the idealized energy conversion rate limit. This is set using a single normal shockwave with a high preshock Mach number (M_{preshock}), as shown in Fig. 7. The idealized energy conversion profile in Fig. 7 is determined from 1-D compressible flow equations with inlet properties determined from a meanline calculation. The supersonic flow is isentropically expanded (see Fig. 7) to a Mach number such that when it is terminated by a shock, the gas pressure is raised to the fixed backpressure. Over a few mean free paths, a single shock can dissipate over 90% of the incoming kinetic energy. However, the shock action yields diminishing returns with increasing M_{preshock} .

For rapidly preheating the gas mixture, further capitalizing on shock action (see Fig. 7) could be a useful technique to reduce the flow path length by increasing the rate of energy conversion. However, for the reaction zone, which demands a specific temperature profile for maximum reaction efficiency, the discontinuity in which the kinetic energy is converted may not necessarily provide the best match to the optimum profile.

3.3 Defining Energy Conversion Metrics. In conventional design practice, loss coefficients are used to quantify efficiency losses and facilitate blade design comparisons. For the turbo-reactor design, the goal is to quantify the amount of incoming

kinetic energy that is converted into enthalpy of the reaction. To quantify the irreversible (entropic) component of energy conversion, the relevant metric is the enthalpy loss coefficient used for conventional machines. However, in this case, we call it the “irreversible energy conversion coefficient” and it is defined for a perfect gas as follows:

$$\zeta_{h,s} = \frac{c_p(T_{\text{exit}} - T_{\text{isentropic}})}{0.5V_1^2} \quad (4)$$

where the state \square_1 is defined at the diffuser inlet and the state \square_{exit} is defined at the end of the vaneless space (see Fig. 8). The value of $T_{\text{isentropic}}$ is the static temperature obtained in an isentropic compression process that ends at the same final pressure as the actual process. The reversible (isentropic) component of the conversion coefficient is defined as

$$\zeta_{h,\text{rev.}} = \frac{c_p(T_{\text{isentropic}} - T_1)}{0.5V_1^2} \quad (5)$$

and the total energy conversion coefficient is defined as

$$\zeta_h = \zeta_{h,s} + \zeta_{h,\text{rev.}} = \frac{c_p(T_{\text{exit}} - T_1)}{0.5V_1^2} \quad (6)$$

4 Numerical Methodology

A combination of unsteady Reynolds-averaged Navier-Stokes (URANS) and high-fidelity large-eddy simulation (LES) strategies

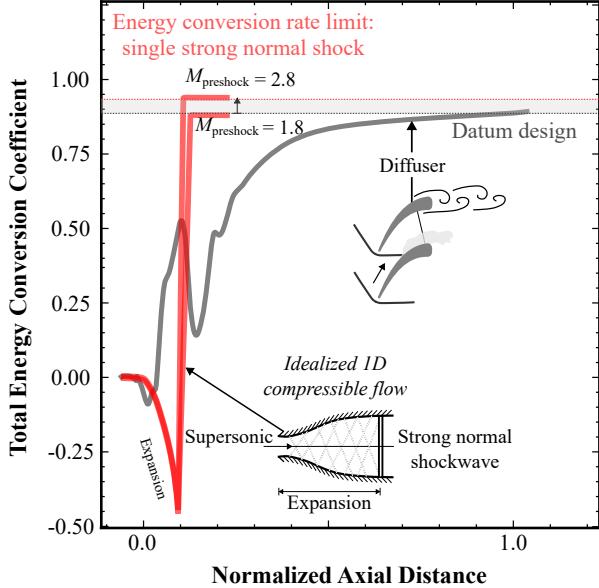


Fig. 7 A comparison of the total energy conversion coefficient (Eq. 5) achieved with the datum design against the idealized limit transferable through a single strong normal shockwave within a 1-D nozzle

has been used to illustrate the mechanisms responsible for the energy conversion process along the flow path. The following section (Sec. 4.1) describes the methodology developed to decompose the flowfield into the various mechanisms.

4.1 Energy Conversion Budget Domain Decomposition.

Metrics. To decompose the energy conversion process into its constituent irreversible mechanisms, as shown in Fig. 8, a similar approach is taken to that of Pullan *et al.* [9] and Zhao *et al.* [10]. The domain is zonally decomposed into 11 spatial subvolumes intimately linked to the physical mechanisms discussed in Sec. 3.1. The Favre-averaged viscous dissipation rate (see Eq. 2) is volume-integrated over each subvolume:

$$\Phi_\tau = \frac{\iiint_V (\bar{\tau}_{ij} + \tau_{ij}^R) \frac{\partial \bar{u}_i}{\partial x_j} dV}{0.5\dot{m}V_1^2} \quad (7)$$

where \mathcal{V} is the subvolume integrated over, \dot{m} is the mass flow rate and V_1 is the diffuser inlet velocity. To quantify the reversible share of energy conversion, the pressure work term in Eq. 2 must be volume-integrated:

$$\Phi_P = \frac{\iiint_V \bar{u}_i \frac{\partial \bar{p}}{\partial x_i} dV}{0.5\dot{m}V_1^2} \quad (8)$$

In conventional loss analysis, the term used to identify sources of loss is the entropy generation rate [11, 12]. However, in the turbo-reactor, since the main objective of the energy conversion breakdown analysis is to identify sources that contribute to the conversion of kinetic energy into energy available for the reaction, the most relevant transport equation is the enthalpy (see Eq. 2) rather than the entropy transport equation. Therefore, Eq. 2 implies that the viscous dissipation rate is the most appropriate metric.

Shock Capturing. The isovolumes associated with a shockwave (see blue isovolumes at the Top of Fig. 10(a)) are identified using the shock function criterion [13]:

$$\sigma = \frac{\mathbf{M} \cdot \nabla p}{|\nabla p|} \quad (9)$$

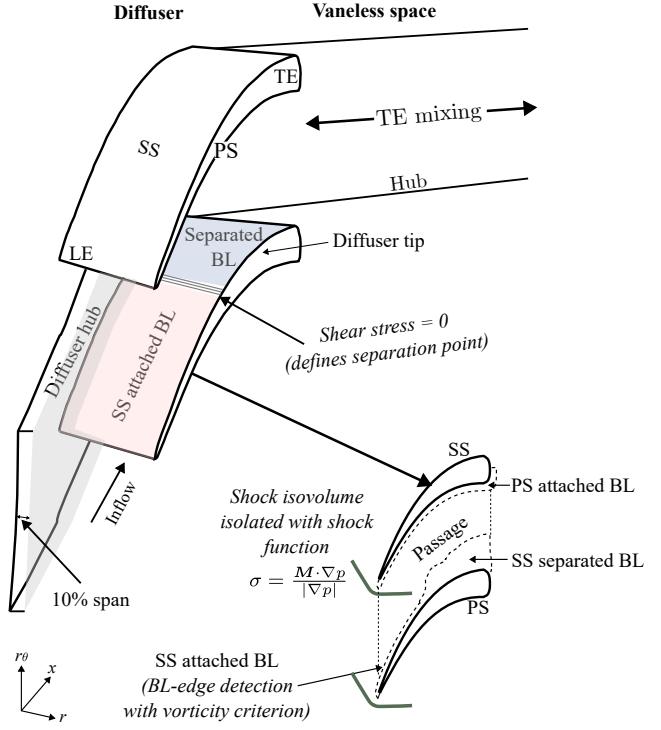


Fig. 8 A schematic illustrating the domain decomposition for the energy conversion breakdown analysis

where \mathbf{M} is the Mach number vector.

Boundary Layer Decomposition. The proceeding analysis distinguishes between an attached and a separated boundary layer. The boundary layer (BL) separation point is defined as the point where the wall shear stress falls to zero (see Fig. 8). The attached boundary layer edge at each streamwise location is determined as the pitchwise station with minimum radial vorticity.

Endwalls. The endwall flow at hub and tip is all the flow within 10% of the respective endwall. This includes the corner flow between the endwalls and the blade surfaces. Tip leakage is not included in this analysis.

Passage Freestream Mixing. The remaining mesh cells not yet classified are associated with passage freestream mixing.

Trailing Edge Mixing. In the vaneless space, all the non-endwall mesh cells immediately downstream of the diffuser trailing edge (TE) are associated with TE mixing (see Fig. 8).

4.2 Numerical Setup. The in-house computational fluid dynamics solver TBLOCK [14, 15] is used for all numerical investigations. The ability of TBLOCK to accurately predict the flowfield where the Reynolds number is similar to that observed in the turbo-reactor has been validated by Rosic *et al.* [16]. All numerical simulations are performed without a reaction model, and the working fluid is modeled as a calorically perfect gas. The perfect gas assumption is reasonable since only a single stage is studied so the thermophysical fluid properties do not change significantly.

LES: Full Stage. The full-stage wall-modeled LES domain contains $226 \times 152 \times 152$ nodes in the axial, spanwise and pitchwise directions within the blade passages (totaling 120 million grid points across the whole domain), leading to an average $y^+ = 4.1$ in the diffuser-vaneless-space system. The Smagorinsky subgrid-scale model with van Driest damping is employed.

URANS: Diffuser and Vaneless Space Domain. For parametric studies on the influence of the Reynolds and Mach number on the energy conversion breakdown, a truncated domain with only diffuser and vaneless space has been used (see Fig. 8).

The inlet boundary condition is taken as the 2-D nonuniform time-averaged field from an unsteady full-stage calculation. It was found that periodic unsteadiness from the passing of the upstream rotor wakes has only a minimal impact on the energy conversion breakdown. Therefore, the diffuser can be safely isolated for this analysis. This is useful because it allows the direct influence of Reynolds and Mach numbers on energy conversion to be decoupled from variations in the upstream flowfield entering the diffuser due to changes in the operating regime.

Table 1 indicates sufficient mesh independence with a domain of $153 \times 103 \times 103$ grid points in the axial, spanwise and pitchwise directions (totaling 18 million grid points), producing an average $y^+ = 5.4$ at the design point. The Spalart–Allmaras turbulence model is employed with wall functions. Due to a high turbulence intensity [1], fully turbulent boundary layers are used.

Table 1 URANS mesh sensitivity for the entropy loss coefficient ζ_s and the irreversible $\zeta_{h,s}$ & total ζ_h energy conversion coefficients

Blade passage grid sizes	ζ_s	$\zeta_{h,s}$	ζ_h
$122 \times 81 \times 81$	0.656	0.632	0.898
$153 \times 103 \times 103^*$	0.652	0.630	0.891
$183 \times 123 \times 123$	0.650	0.630	0.891

Achieving Entropy Balance. Since turbulent dissipative mechanisms are being investigated in this work, the entropy balance across the domain is also assessed [11, 17, 18] to evaluate mesh convergence. For URANS over 90% of the entropy flux across the domain was modeled by the entropy generation rate, but less than 50% was resolved-modeled with LES due to sensitivity to numerical dissipation. Therefore, URANS is selected to determine the energy conversion breakdown because in this case, the modeled viscous dissipation almost fully determines the turbulent dissipation distribution. Although this introduces uncertainty due to turbulence modeling, it provides an efficient approach to gaining a deeper understanding into the energy conversion budget.

5 Energy Conversion Mechanisms

To enable fine-tuning of the energy conversion profile, it is important to understand the energy transformation processing mechanisms. This will motivate aerothermal design optimization to better match the optimum temperature profile for the chemistry (see Sec. 3.1). For example, the diffuser geometry could be modified accordingly, and the vaneless space length can be varied stage by stage (see Figs. 3 & 4).

5.1 Aerodynamically-Tuned: Energy Transfer Process. Before the energy conversion process is illustrated in detail using numerical simulations, it is instructive to see how the aerodynamic losses (*i.e.*, the entropy rise) vary across the stator-rotor system, where mechanical energy is imparted into the flow. Figure 9 highlights a clear distinction between the rotor-stator system and diffuser-vaneless-space system. While the latter is designed to amplify losses to chemically tune the reaction, the former is aerodynamically tuned, which is more in accordance with conventional turbomachinery design practice. However, while conventional design focuses on increasing efficiency across the rotor and throughflow capacity per unit frontal area of the machine, the design of the turbo-reactor stator-rotor system is more concerned with developing a high work coefficient as well as throughflow capacity. This directly translates into reducing flow deviation and blockage. This,

of course, is strongly correlated with reducing the entropy rise (see Fig. 9). However, it is not necessarily the absolute minimum entropy rise that corresponds to the maximum power input.

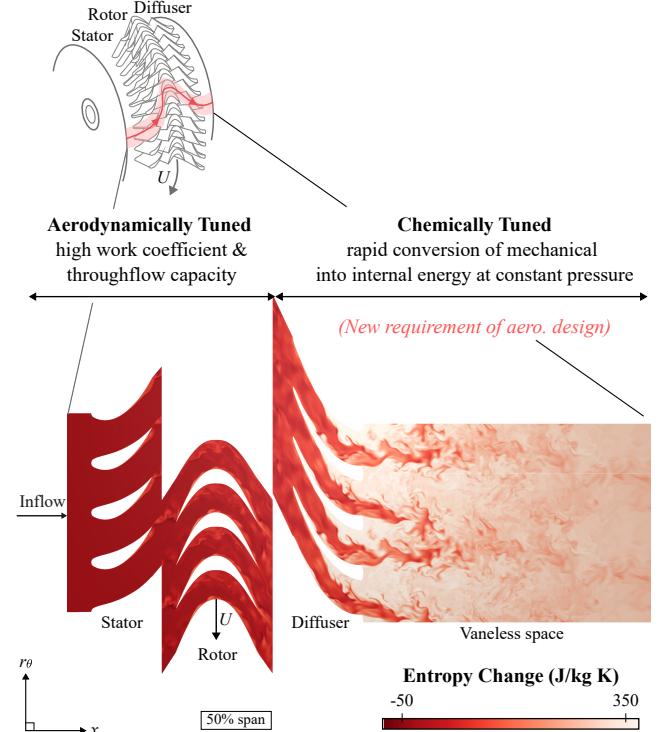


Fig. 9 Single stage instantaneous entropy change (LES)

5.2 Chemically-Tuned: Energy Conversion Process. Figure 10(a) shows the energy conversion process in the diffuser-vaneless system. Irreversibilities dominate over reversible pressure work, accounting for a 65% share (see Fig. 10(b)). This is primarily a consequence of turbulent rather than molecular viscosity. The diffuser blade row accounts for 55% of the total energy conversion. Within $10\times$ diffuser TE thicknesses (t_{TE}), the energy conversion process is almost complete and, strikingly, $\sim 90\%$ ($\zeta_h \approx 0.9$) of the incoming kinetic energy is dissipated in under $100\mu\text{s}$ (see Fig. 10(b)). The loss coefficient is approximately 10 times that developed in a compressor. Uniform conditions are quickly established just downstream of the diffuser TE, which is crucial for optimal reaction performance.

The remaining 35% of energy conversion is composed of fairly equal contributions from the reversible pressure gain component of the shock-induced temperature rise and isentropic diffusion of the flow (see Fig. 10(b)). Energy conversion across the thin shock layer has two distinct contributions (see Eq. 2).

- Reversible pressure work (accounted for by $\bar{u}_i \frac{\partial \bar{p}}{\partial x_i}$).
- Irreversibilities (accounted for by $(\bar{\tau}_{ij} + \tau_{ij}^R) \frac{\partial \bar{u}_i}{\partial x_j}$).

At low Mach numbers (M), the former dominates, but as M increases, the latter begins to dominate [19]. It will be shown in Sec. 5.3 that the temperature change due to entropy produced in the shock system is almost negligible. Therefore, the overall contribution of the shock system is about 20% (see Fig. 10(b)), primarily due to reversible pressure work rather than entropy.

Figure 10(a) suggests that the dominant impact of the shock system on the irreversible energy conversion rate is an indirect rather than direct effect. The impact of the shock system is the result of the shockwave boundary layer interaction (SWBLI). Shock impingement separates the suction surface (SS) boundary layer at

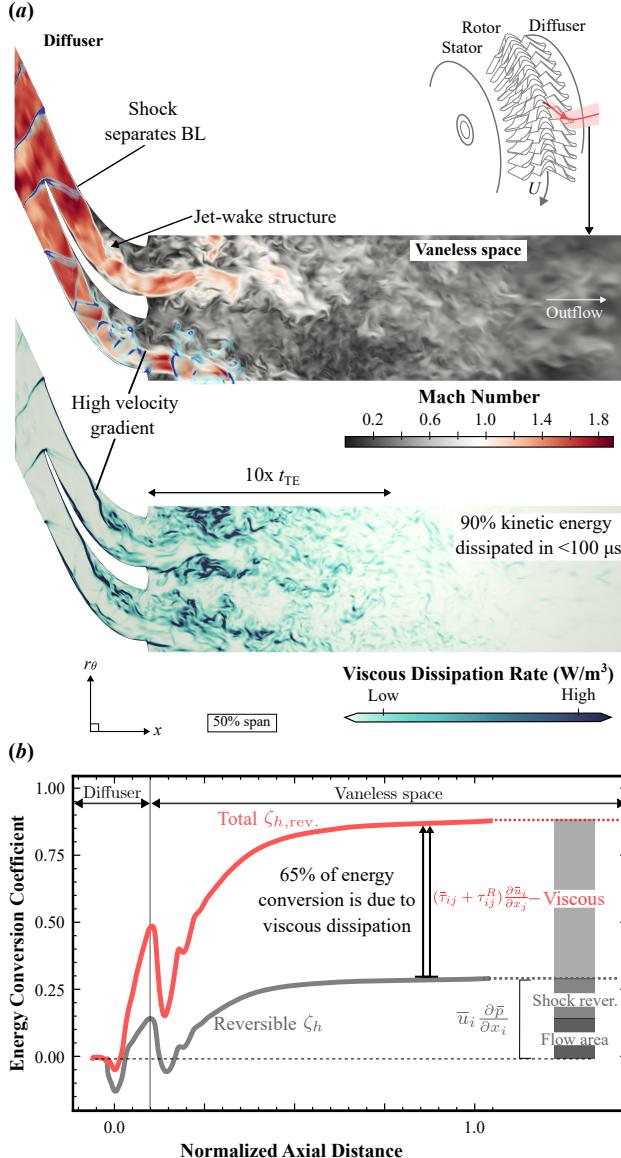


Fig. 10 (a) Instantaneous Mach number with superimposed numerical schlieren in blue (first) and viscous dissipation (second), and (b) time- and mass-averaged total (ζ_h) and reversible ($\zeta_{h,rev.}$) energy conversion coefficient (LES)

approximately 50% of the chord. The high-entropy boundary layer unravels and spreads, mixing out the hot boundary layer fluid into the freestream. At the interface between the separated backflow region and the freestream jet above it, a large velocity gradient is generated (see Fig. 10(a)). This leads to intense deformation shear work ($(\bar{\tau}_{ij} + \tau_{ij}^R) \frac{\partial \bar{u}_i}{\partial x_j}$) at the interface (see Fig. 10(a)). Furthermore, Fig. 10(a) (Top) shows, in blue, the footprint of a shock diamond within the overexpanded jet that exits the diffuser passage. This has important implications for TE mixing [20].

Despite the transonic flow regime and the thick TE occupying $t_{TE}/w \approx 30\%$ of the blade pitch w , surprisingly, Fig. 11 implies a minimal impact of the base pressure coefficient C_{pb} on TE loss. It is calculated as $C_{pb} \approx -0.05$ around the TE circle, which is several times smaller in magnitude than that of a transonic turbine ($C_{pb} \approx -0.3$) [20]. This is because the SS separation sets the pressure level around the TE (see Fig. 11(b)).

Downstream, most of the mechanical energy dissipation is due to the mixing of two bulk streams—the jet and the separated wake

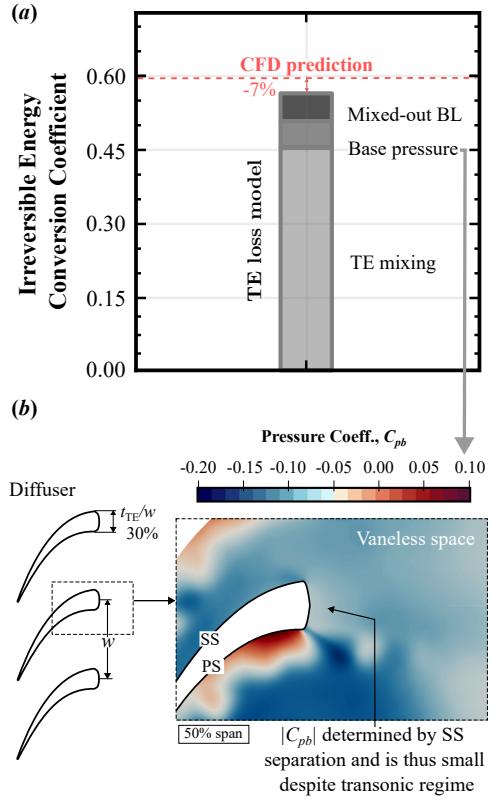


Fig. 11 (a) Irreversible energy conversion coefficient predicted by a modified Denton TE loss model [12, 21] and (b) time-averaged pressure coefficient ($C_{pb} = \frac{p-p_{ref}}{p_{01}-p_{ref}}$, where p_{ref} is the average pressure PS and SS before the TE circle)

(see Fig. 10(a))—which have widely different stagnation pressures. The entropy generated in velocity equilibration to restore uniform flow is directly related to the difference in stagnation pressures between the jet and the wake [12]. Since this is large (see Fig. 10(a)), the downstream mixing intensifies. To model this mixing, a modified version of Denton's TE loss model [12, 21] shown in Fig. 11(a) provides a fairly accurate estimate of the energy conversion despite the model being extrapolated substantially beyond its typical application range. Remarkably, the irreversible energy conversion coefficient is only underestimated by 7%.

5.3 Global Energy Conversion Breakdown Analysis. Based on the domain decomposition introduced in Sec. 4.1, Fig. 12 shows the streamwise evolution of the irreversible energy conversion breakdown for a typical design and boundary conditions. Most importantly, it is clear that turbulent dissipation generated downstream of the diffuser TE is the dominant mechanism, accounting for 38% of the total. This process almost reaches its maximum proportion within $1C_x$ (see Fig. 12). Because the SS boundary layer separates far upstream of the TE (see Fig. 10(a)), a thick TE may not be necessary to encourage rapid mixing (although it is beneficial from a thermomechanical perspective).

There are three further conclusions to be drawn from Fig. 12. First, the pressure surface (PS) attached boundary layer energy conversion dominates over the SS BL because it is attached for longer. The SS boundary layer separates earlier (see Fig. 10(a)) and is therefore the most significant contribution within the diffuser passage. Second, in both the diffuser passage and vaneless space, the tip flow near the endwall generates more entropy than at the hub because the Mach number is higher and more mass flow is concentrated (which is implied by radial equilibrium). Finally, the viscous dissipation within the shock system is relatively small, as it mainly comprises relatively weak oblique shocks. This is

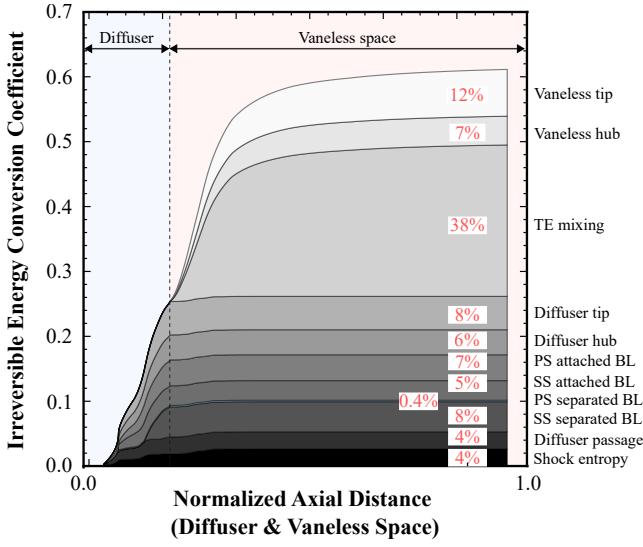


Fig. 12 Streamwise evolution of the irreversible energy conversion coefficient breakdown

consistent with the fact that the entropy change across a shock is only of third order in shock strength.

5.4 Mach and Reynolds Number Effects. For a perfect gas, three nondimensional flow parameters govern the energy conversion process in the diffuser: the Reynolds number, Mach number and diffuser inlet flow angle. This section explores Reynolds and Mach effects over an engine-relevant range using 4 \times samples to cover each nondimensional group. This representative range covers the variability of nondimensional groups across the multistage environment [5]. Due to the unique incidence condition [22], the inlet flow angle is not constant as the diffuser inlet Mach number varies; however, it only varies by $\sim 1^\circ$.

As the diffuser inlet Mach number (M) increases for a fixed Reynolds number, Fig. 13 shows that the total energy conversion coefficient increases, but the irreversible contribution decreases. The total increases with Mach number for two reasons. First, there is a greater contribution of the reversible pressure work through the shock system (see Fig. 13) due to an increased pressure ratio. There is a considerable difference in the shock system topology due to non-linear variations in supersonic expansion within the passage across the range. Second, the passage isentropic flow diffusion increases due to changes in the effective flow area.

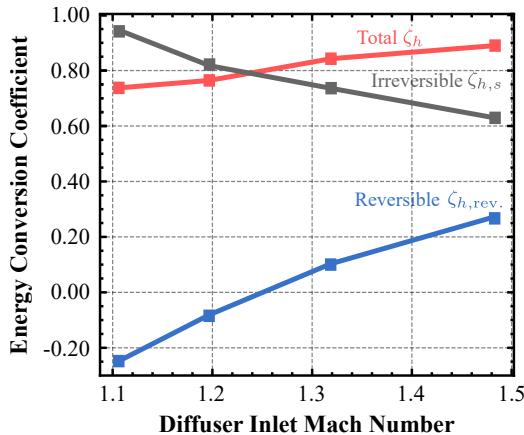


Fig. 13 Energy conversion coefficients as a function of diffuser inlet Mach number for a fixed Reynolds number

Figure 14(a) indicates that although the overall distribution of

irreversible energy conversion mechanisms does not change dramatically, its overall magnitude decreases by 33% with increasing Mach number. A key factor is the reduction in normalized dissipation in the attached blade and endwall boundary layers. This is likely to be a consequence of two factors. First, the dissipation coefficient C_d is known to decrease with Mach number (for a fixed Reynolds number) [23], but only marginally. Second, it was found that at higher Mach numbers, the flow acceleration in the passage is reduced, changing the state of the boundary layer and leading to a lower C_d . Furthermore, the diffuser exit flow angle α was found to decrease by 13° over the range. This directly translates to lower TE mixing ($\zeta_{h,s} \propto 1/(w \cos \alpha)$), as seen in Fig. 14(a), as a result of reduced sudden flow expansion.

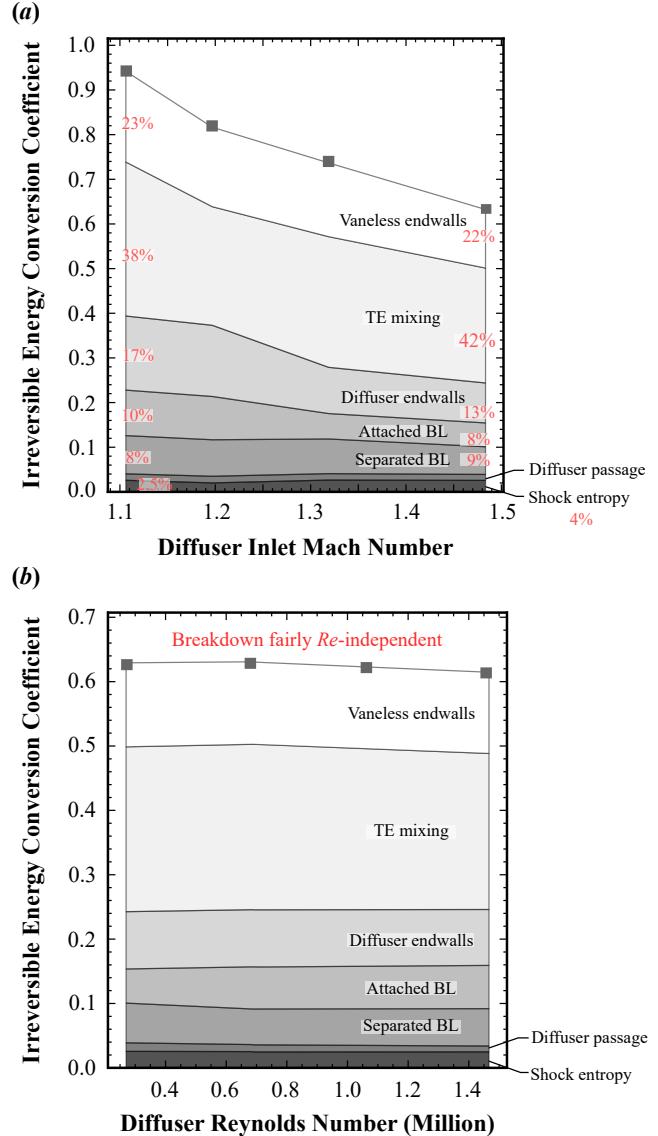


Fig. 14 The irreversible energy conversion coefficient as a function of: (a) diffuser inlet Mach number and (b) Reynolds number ($Re = \frac{\rho_1 V_1 b_{true}}{\mu}$) based on the diffuser inlet conditions ($\rho_1 V_1$) and its true chord length (b_{true})

Compared to the Mach number, the consequence of Reynolds number (Re) variability is negligible and the energy conversion breakdown is approximately independent of the Reynolds number over a range from 0.4 million to 1.4 million, as shown in Fig. 14(b). There is a minor drop in the irreversible energy conversion coefficient at higher Reynolds numbers, since dissipation is

inversely related to Re and the distribution of mechanisms changes marginally. Reynolds number independence is perhaps expected, since Re is above the critical Reynolds number ($Re_{crit} \approx 100,000$) and turbulence intensity at the diffuser inlet is high ($\geq 10\%$).

5.5 Compact Multistage Architecture. For preheating the nonreacting gas mixture before reaction activation, temperature profiling is unnecessary, and a “compact” turbo-reactor stage can be adopted (see Fig. 3). Since the vaneless mixing space is absent, the flow can maintain a higher Mach number when it enters stages 2 to 4 relative to the first stage.

Figure 15 illustrates that despite the removal of the vaneless mixing space and diffuser, it is still possible to dissipate a large amount of mechanical energy within each stage, albeit through different mechanisms. As expected, before the final diffuser, TE mixing no longer plays a significant role in the energy conversion process. Instead, viscous dissipation within the blade surface and endwall boundary layers dominates. This is driven by an increasing freestream velocity across the multistage environment.

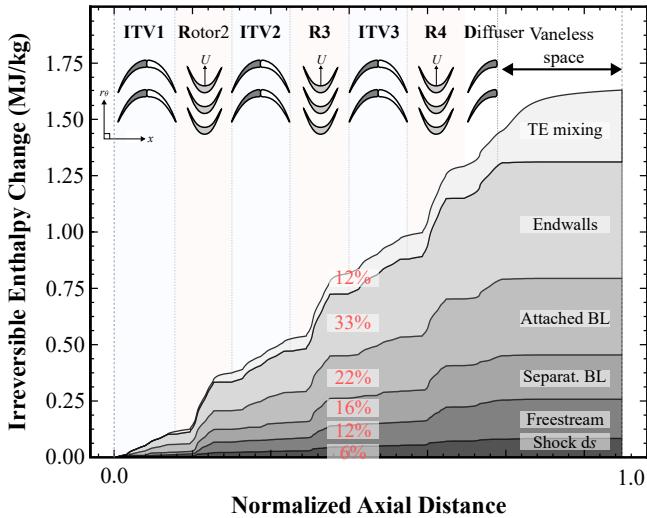


Fig. 15 Streamwise breakdown of the irreversible component of enthalpy change across the multistage compact turbo-reactor architecture

6 Summary and Conclusions

This paper has explained that the turbo-reactor can be designed as a “chemically tuned” device in which the bladed path is tailored to optimize the reaction dynamics. For the first time in turbomachinery, aerodynamic losses can be leveraged to control the rate of energy conversion into the reaction and thus design a reaction-efficient temperature profile. This can be achieved by optimally balancing dissipation against the endothermic sink term. By introducing new vectors of energy quality for turbomachines, chemical exergy and reaction efficiency, the thermodynamic implications of converting electricity to heat have been clarified.

To enable optimization of the aerodynamic design to better match the ideal energy supply profile into the chemical reaction, it is crucial to explore the mechanisms responsible for the energy conversion process. A series of numerical investigations have been performed to examine the breakdown of energy conversion mechanisms. Several conclusions have been drawn.

- (1) Turbulent viscous dissipation downstream of the diffuser TE is the dominant mechanism of energy conversion (at 25% of the total). This is a crucial process for producing a uniform composition distribution that favors a more efficient reaction.
- (2) The total contribution of the shock system was around 20%, mainly due to a reversible pressure work mechanism rather

than entropy rise. The dominant irreversible contribution of the shock system is indirect through SWBLI. This separates the boundary layer early in the passage and initiates the strong mixing process that continues downstream of the TE.

- (3) The energy conversion breakdown has been explored over a range of Mach and Reynolds numbers. Over a wide range of flow regimes it is possible to dissipate a large fraction of the incoming kinetic energy ($\geq 60\%$). As expected, the Mach number had a more significant influence. This means that in a multistage environment, the normalized rate of energy conversion will fall with the decreasing Mach number towards the rear of the machine. This must be taken into account when designing a reaction-efficient temperature profile.
- (4) For the compact stage design, the energy conversion rate remains high despite eliminating the interstage vaneless mixing space. This is partly a consequence of elevated skin friction due to increased flow speeds.

We hope that this paper opened an entirely new field of research into tuning the aero-thermal design of turbomachines to control and optimize chemical transformations along the bladed flow path through careful temperature profile design.

Acknowledgments

The authors thank N. Karefyllidis for providing the stage dataset. The authors appreciate the fruitful discussions with N. A. Cumpsty. The computational resources used were provided by [ARCHER2](#).

Nomenclature

Roman Letters

b_{true}	= True diffuser chord length, m
c_p	= Isobaric heat capacity, $J \text{kg}^{-1} \text{K}^{-1}$
C_x	= Diffuser axial chord length, m
h_0	= Total enthalpy, J kg^{-1}
p	= Static pressure, bar
t_{TE}	= Diffuser trailing edge thickness, m
T	= Static temperature, K
u_i	= Velocity vector, m s^{-1}
U	= Mean blade speed, m s^{-1}
V_1	= Diffuser inlet velocity, m s^{-1}
w	= Diffuser blade pitch, m

Greek Letters

ζ_h	= Total energy conversion coefficient
$\zeta_{h,rev.}$	= Reversible (isentropic) energy conversion coefficient
$\zeta_{h,s}$	= Irreversible (entropic) energy conversion coefficient
μ	= Dynamic molecular viscosity, Pa s
ρ	= Density, kg m^{-3}
σ	= Shock function, $\frac{\mathbf{M} \cdot \nabla p}{ \nabla p }$
$\bar{\tau}_{ij}$	= Mean flow shear stress tensor, Pa
τ_{ij}^R	= Reynolds stress tensor, Pa

Dimensionless Groups

M	= Mach number
Re	= Reynolds number

Superscripts

\square'	= Fluctuating component
$\overline{\square}$	= Time averaged

Acronyms

- BL = Boundary layer
- ITV' = Integrated turning vane
- LES = Large-eddy simulation
- LE/TE = Leading/trailing edge
- PS/SS = Pressure/suction surface
- SWBLI = Shockwave boundary layer interaction

Appendix A: Exergy

Total specific exergy is defined as follows:

$$b_{\text{ex}}^{\text{tot}} = b_{\text{ex}}^{T\text{p}} + b_{\text{ex}}^{\text{ch}} \quad (\text{A1})$$

The physical (thermomechanical) component of the total exergy is:

$$b_{\text{ex}}^{T\text{p}} = (h_0 - h_{0,\text{ref}}) - T_{\text{ref}}(s - s_{\text{ref}}) \quad (\text{A2})$$

where the subscript \square_{ref} is the thermochemical state of the reference environment [24], h_0 is the stagnation enthalpy and s is the entropy. The chemical exergy is:

$$b_{\text{ex}}^{\text{ch}} = \sum_{k=1}^{N_{\text{spe}}} Y_k b_{\text{ex},k}^{\text{ch,ref}} + RT_{\text{ref}} \sum_{k=1}^{N_{\text{spe}}} X_k \ln X_k \quad (\text{A3})$$

where X_k are the species mole fraction, Y_k are the mass fractions, and $b_{\text{ex},k}^{\text{ch,ref}}$ is the chemical exergy of each species.

Chemical exergy is the result of the chemical disequilibrium of a species with the reference environment. This is calculated using the Gibbs function defined for a chemical reaction between the substance in question and the reference environment. In theory, useful work can be obtained from a chemical resource (species) by promoting a reaction until a chemical equilibrium is reached with the reference environment. The chemical exergy of various chemicals, fuels and materials can be found tabulated by Szargut *et al.* [24]. Note that for most hydrocarbon fuels, the chemical exergy is very similar to the heating value of the fuel.

References

- [1] Rubini, D., Xu, L., Rosic, B., and Johannesdahl, H., 2021, "A New Turbomachine for Clean and Sustainable Hydrocarbon Cracking," *ASME J. Eng. Gas Turbines Power*, **144**(2), p. 021024.
- [2] Rubini, D., Karefyllidis, N., Xu, L., Rosic, B., and Johannesdahl, H., 2022, "A New Robust Regenerative Turbo-Reactor Concept for Clean Hydrocarbon Cracking," *J. Glob. Power Propuls. Soc.*, **6**, pp. 135–150.
- [3] Rubini, D., Karefyllidis, N., Xu, L., Rosic, B., and Johannesdahl, H., 2022, "Accelerating the Development of a New Turbomachinery Concept in an Environment With Limited Resources and Experimental Data: Challenges," *Volume 10D: Turbomachinery — Multidisciplinary Design Approaches, Optimization, and Uncertainty Quantification; Turbomachinery General Interest; Unsteady Flows in Turbomachinery*, Rotterdam, Netherlands, 2022, p. V10DT36A001, doi: [10.1115/GT2022-80698](https://doi.org/10.1115/GT2022-80698).
- [4] Karefyllidis, N., Rubini, D., Rosic, B., Xu, L., and Purola, V.-M., 2023, "A Novel Axial Energy-Imparting Turbomachine for High-Enthalpy Gas Heating: Robustness of the Aerodynamic Design," *ASME J. Turbomach.*, **146**(3), p. 031005.
- [5] Rubini, D., Karefyllidis, N., Rosic, B., Xu, L., and Nauha, E., 2024, "Decarbonisation of high-temperature endothermic chemical reaction processes using a novel turbomachine: robustness of the concept to feed variability," *J. Glob. Power Propuls. Soc.*, **8**, pp. 111–126.
- [6] Tijani, M. E. H., Zondag, H., and Van Delft, Y., 2022, "Review of Electric Cracking of Hydrocarbons," *ACS Sustain. Chem. Eng.*, **10**(49), pp. 16070–16089.
- [7] Dickens, T. and Day, I., 2010, "The Design of Highly Loaded Axial Compressors," *ASME J. Turbomach.*, **133**(3), p. 031007.
- [8] Poinsot, T. and Veynante, D., 2005, *Theoretical and Numerical Combustion*, RT Edwards, Inc.
- [9] Pullan, G., Denton, J., and Curtis, E., 2004, "Improving the Performance of a Turbine With Low Aspect Ratio Stators by Aft-Loading," *ASME J. Turbomach.*, **126**(3), pp. 492–499.
- [10] Zhao, Y. and Sandberg, R. D., 2020, "Using a New Entropy Loss Analysis to Assess the Accuracy of RANS Predictions of an High-Pressure Turbine Vane," *ASME J. Turbomach.*, **142**(8), p. 081008.

- [11] Zlatinov, M. B., Sooi Tan, C., Montgomery, M., Islam, T., and Harris, M., 2012, "Turbine Hub and Shroud Sealing Flow Loss Mechanisms," *ASME J. Turbomach.*, **134**(6), p. 061027.
- [12] Denton, J. D., 1993, "The 1993 IGTI Scholar Lecture: Loss Mechanisms in Turbomachines," *ASME J. Turbomach.*, **115**(4), pp. 621–656.
- [13] Lovely, D. and Haines, R., 1999, "Shock Detection from Computational Fluid Dynamics Results," *14th computational fluid dynamics conference*, Norfolk, VA, U.S.A., 1999, p. 3285, doi: <https://doi.org/10.2514/6.1999-3285>.
- [14] Denton, J. D., 1983, "An Improved Time-Marching Method for Turbomachinery Flow Calculation," *ASME J. Eng. Gas Turbines Power*, **105**(3), pp. 514–521.
- [15] Denton, J. D., 1986, "The Use of a Distributed Body Force to Simulate Viscous Effects in 3D Flow Calculations," *Volume 1: Turbomachinery, Turbo Expo: Power for Land, Sea, and Air*, Vol. Volume 1: Turbomachinery, Dusseldorf, West Germany, 1986, p. V001T01A058, doi: [10.1115/86-GT-144](https://doi.org/10.1115/86-GT-144).
- [16] Rosic, B., Denton, J. D., and Pullan, G., 2005, "The Importance of Shroud Leakage Modeling in Multistage Turbine Flow Calculations," *ASME J. Turbomach.*, **128**(4), pp. 699–707.
- [17] Przytarski, P. J. and Wheeler, A. P. S., 2021, "Accurate Prediction of Loss Using High Fidelity Methods," *ASME J. Turbomach.*, **143**(3), p. 031008.
- [18] Jardine, L. and Miller, R., 2021, "The Effect of Heat Transfer on Turbine Performance," *ASME J. Turbomach.*, pp. 1–34.
- [19] Greitzer, E. M., Tan, C. S., and Graf, M. B., 2007, *Internal Flow: Concepts and Applications*, 3rd ed., Cambridge University Press, Cambridge.
- [20] Denton, J. D. and Xu, L., 1990, "The Trailing Edge Loss of Transonic Turbine Blades," *ASME J. Turbomach.*, **112**(2), pp. 277–285.
- [21] Senior, A. C. and Miller, R. J., 2023, "A Data-Centric Approach to Loss Mechanisms," *ASME J. Turbomach.*, **146**(4), p. 041007.
- [22] Cumpsty, N. A., 2004, *Compressor Aerodynamics*, Krieger Publishing, Florida.
- [23] Tosto, F., Wheeler, A., and Pini, M., 2024, "Investigation of non-ideal effects in compressible boundary layers of dense vapors through direct numerical simulations," *Phys. Fluids*, **36**(1), p. 016145.
- [24] Szargut, J., Morris, D., and Steward, F., 1988, *Exergy Analysis of Thermal, Chemical, and Metallurgical Processes*, Hemisphere Publishing Corporation.