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A REVIEW OF WAVE ROTOR TECHNOLOGY AND ITS APPLICATIONS

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ABSTRACT

The objective of this paper is to provide a succinct review of past and current research in developing wave rotor technology. This technology has shown unique capabilities to enhance the performance and operating characteristics of a variety of engines and machinery utilizing thermodynamic cycles. Although there have been numerous efforts in the past dealing with this novel concept, this technology is not yet widely used and barely known to engineers. Here, an attempt is made to summarize both the previously reported work in the literature and ongoing efforts around the world. The paper covers a wide range of wave rotor applications including the early attempts to use wave rotors, its successful commercialization as supercharges for car engines, research and development for gas turbine topping, and other developments. The review also pays close attention to more recent efforts: utilization of such devices in pressure-gain combustors, ultra-micro gas turbines, and water refrigeration systems, highlighting possible further efforts on this topic. Observations and lessons learnt from experimental studies, numerical simulations, analytical approaches, and other design and analysis tools are presented.

Keywords: *wave rotor, dynamic pressure exchanger, energy exchanger, gas turbine, shock wave, expansion wave, unsteady flow*

INTRODUCTION

Oscillatory and pulsatile fluid motion has been neatly utilized in nature, yet is comparatively poorly studied by engineers despite the invention of cyclically operating engines and machines. The potential for utilizing unsteady flows has been recognized since the early twentieth century, but neglected as long as substantive improvements could be made to conceptually simple steady-flow or semi-static devices. Also, the inherent non-linearity of large-amplitude wave phenomena in compressible fluids and unusual geometry of unsteady flow devices necessitates detailed calculations, which until recently were too laborious or expensive or imprecise. By understanding and exploiting complex unsteady flows, a quantum increase in engine performance is possible. Often it has been feasible to simplify the hardware of engines, making them less costly, more responsive, and more durable by employing unsteady processes.

Shock tubes, shock tunnels, pulse combustors, pulse detonation engines, and wave rotors are a few examples of unsteady-flow devices. The basic concept underlying these devices is the transfer of energy by pressure waves. By generating compression and expansion waves in appropriate geometries, wave machines can transfer energy directly between different fluids without using mechanical components such as pistons or vaned impellers. In fact, these devices properly represent applications of classical, unsteady, one-dimensional, compressible flow theory. The major benefit of these unsteady-flow machines is their potential to generate much greater pressure rises than those obtained in steady-state flow devices [1-2]. Furthermore, shock wave compression is a relatively efficient process as shown in Fig 1. where shock isentropic efficiency η_{Shock} (red) is compared with compressor isentropic efficiency $\eta_{Compressor}$ (green) and diffuser isentropic efficiency $\eta_{Diffuser}$ (blue). Figure 1 shows variations of these parameters as functions of the pressure gain p_2/p_1 obtained by a

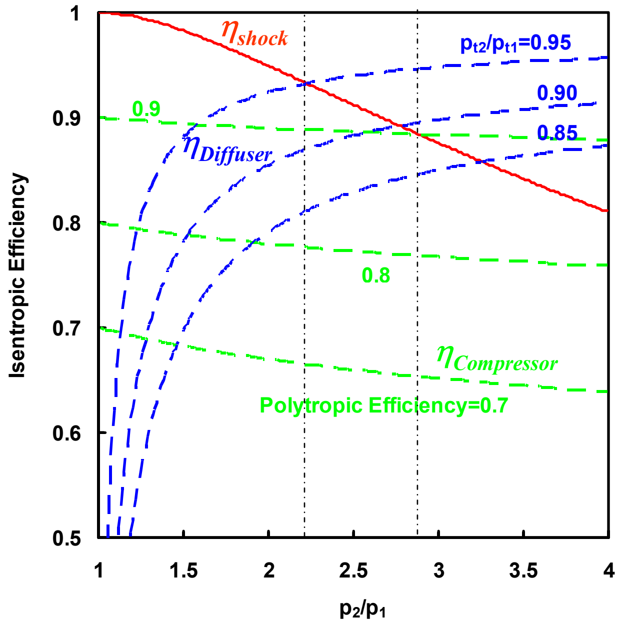

Figure 1: Shock wave, compressor, and diffuser isentropic efficiencies as function of pressure gain

moving shock wave in a frictionless channel, by a compressor with different values of polytropic efficiencies, and by a diffuser with different values of total pressure drop across the diffuser expressed by p_{t2}/p_{t1} , respectively. The comparison reveals that for the same pressure gain p_2/p_1, the ideal shock compression efficiency may far exceed the efficiency obtained by a decelerating diffuser or a compressor. For example, it is seen that for a pressure gain of up to $p_2/p_1=2.2$, η_{Shock} is greater than about 93% and thus is greater than those of typical diffusers and compressors. Therefore a gain in cycle performance can be expected when a compressor (or a diffuser) is replaced by an unsteady-flow device utilizing shock waves. Flow friction effects would lower the efficiency of wave devices and reduces their efficiency advantage (not shown in Fig. 1), but the relative advantage is expected to persist.

WAVE ROTOR MACHINES

The essential feature of wave rotors is an array of channels arranged around the axis of a cylindrical drum. As schematically shown in Fig. 2, the drum rotates between two end plates each of which has a few ports or manifolds, controlling the fluid flow through the channels. The number of ports and their positions vary for different applications. By carefully selecting their locations and the widths to generate and utilize wave processes, a significant and efficient transfer of energy can be obtained between flows in the connected ducts. Through rotation, the channel ends are periodically exposed to the ports located on the stationary end plates initiating compression and expansion waves within the wave rotor channels. Thus, pressure is exchanged dynamically between fluids by utilizing unsteady pressure waves. Therefore, unlike a steady-flow turbomachine which either compresses or expands the fluid, the wave rotor accomplishes both

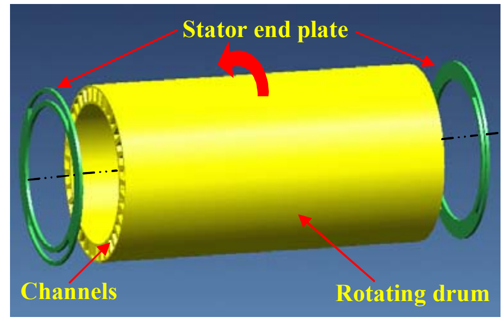

Figure 2: Schematic configuration of a typical wave rotor

compression and expansion within a single component. To minimize leakage, the gap between the end plates and the rotor has to be very small or the end plates with sealing material could contact the rotor. An inverted design with stationary rotor and rotating ports is also possible [3]. Such a configuration may be preferred for laboratory investigations because it easily enables flow measurement in the channels where the important dynamic interactions take place. However, this arrangement rarely seems to be convenient for commercial purposes [4]. The rotor may be gear or belt driven or preferably direct driven by an electrical motor (not shown). The power required to keep the rotor at a correctly designed speed is negligible [4, 5]. It only needs to overcome rotor windage and friction in the bearings and contact sealing if used. Alternatively, rotors can be made self-driving. This configuration, known as the “free-running rotor”, can drive itself by using the momentum of the flow to rotate the rotor [6, 7].

In wave rotor machines, two basic fluid-exchange processes usually happen at least once per revolution of the rotor: the high-pressure process (charging process) and the low-pressure process (scavenging process). In the high-pressure process, compression waves transfer the energy directly from a fluid at a higher pressure (driver fluid) to another fluid at a lower pressure (driven fluid). In the low-pressure process the driver fluid is scavenged from the rotor channels, using expansion waves. Generation of expansion waves allows ingestion of a fresh low-pressure fluid into the rotor channels.

There are several important advantages of wave rotor machines. Their rotational speed is low compared with turbomachines, which results in low material stresses. But they can respond on the timescale of pressure waves, with no rotor inertial lag. From a mechanical point of view, their geometries can be simpler than those of turbomachines. Therefore, they can be manufactured relatively inexpensively. Also, the rotor channels are less prone to erosion damage than the blades of turbomachines. This is mainly due to the lower velocity of the working fluid in the channels, which is about one-third of what is typical within turbomachines [4]. Another important advantage of wave rotors is their self-cooling capabilities. They are naturally cooled by the fresh cold fluid ingested by the rotor. Therefore, applied to a heat engine, the rotor channels pass through both cool air and hot gas flow in the cycle at least once per rotor revolution. As a result, the rotor material

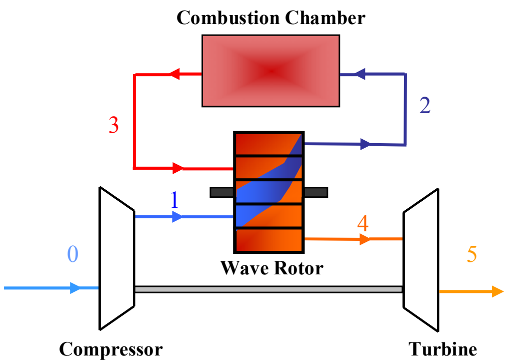

Figure 3: Schematic of a gas turbine topped by a through-flow four-port wave rotor

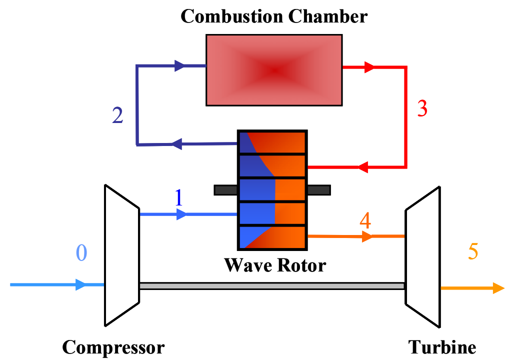

Figure 4: Schematic of a gas turbine topped by a reverse-flow four-port wave rotor

temperature is always maintained between the temperature of the cool air which is being compressed, and the hot gas which is being expanded.

Despite very attractive features, several challenges have impeded the vast commercial appearance of wave rotors in some applications though numerous research efforts have been carried out during the past century. Besides unusual flow complexity and anticipated off-design problems and uncertainties about the selection of the best wave rotor configuration for a particular application, the obstacles have been mainly of a mechanical nature, like sealing and thermal expansion issues, as mentioned throughout this study. However, due to the recent energy crises, technology improvement, and economic reasons, new desires for wave rotor technology have been stimulated.

FOUR-PORT WAVE ROTOR EXAMPLES

A variety of wave rotor configurations have been developed for different applications. The number and azimuthal location of the wave rotor ports along with heat addition schemes distinguish them for different purposes. As will be shown in the next section, four-port configurations have been mainly used as superchargers for internal combustion engines. Three port wave rotors have been employed in pressure dividers and pressure equalizers in which the pressures of different fluids are increased or reduced. Two-port, four-port, five-port, and nine-port wave rotors have been extensively investigated for gas turbine engine topping applications. As an application of the current interest, a four-port wave rotor integrated into a gas turbine cycle is briefly discussed below to illustrate wave rotor operation and options.

Figure 3 shows a schematic of a gas turbine cycle using a four-port wave rotor. Following the flow path shown in Fig. 3, air from the compressor enters the wave rotor (state 1) and is further compressed inside the wave rotor channels. After the additional compression of the air in the wave rotor, it discharges into the combustion chamber (state 2). Thus, combustion takes place at a higher pressure and temperature than in the baseline engine. The hot gas leaving the combustion chamber (state 3) enters the wave rotor and compresses the air received from the compressor (state 1). To provide the energy transfer to compress the air, the burned gas expands and is afterward scavenged toward the turbine (state 4). Due to the pre-expansion in the wave rotor, the burned gas enters the turbine with a lower temperature than that of the combustor exit. However, the gas pressure is still higher than the compressor exit pressure by the pressure gain obtained in the wave rotor. The turbine inlet total pressure is typically 15 to 20% higher than pressure of the air delivered by the compressor [8]. This pressure gain is in contrast to the untopped engine where the turbine inlet pressure is always lower than the compressor discharge pressure due to the pressure loss across the combustion chamber. As a result of the wave rotor pressure gain, more work can be extracted from the turbine increasing overall engine thermal efficiency and specific work. Finally, the channels are re-connected to the compressor outlet, allowing fresh pre-compressed air to flow into the wave rotor channels and the cycle repeats.

In the above described wave rotor, both gas and air inlet ports are located on one side of the rotor while and the outlet ports are located on the other side of the rotor. This configuration is know as the through-flow (TF) wave rotor in the literature. Alternatively, another type of wave rotor has been designed where the fresh air enters and exits at the same end of the rotor (air casing) while the burned gas enters and exits the rotor at the other end (gas casing). This configuration is called reverse-flow (RF) wave rotor, shown in Fig. 4. These two configurations may provide identical topping and overall performance enhancement, but they differ substantially in their internal processes. In the TF four-port wave rotor, both hot gas and relatively cold air traverse the full length of the rotor, keeping the wall at a relatively uniform intermediate temperature. This self-cooling feature of TF wave rotors has prompted interest in them for gas turbine engine topping applications where gas temperatures are high. The RF configuration does not inherently result in such a self-cooled rotor. The cold air never reaches the other end of the rotor, as seen from Fig. 4. As a result, the air side of the rotor is relatively cool while the gas side of the rotor is relatively hot. Thus, the RF configurations have been mostly used in the relatively low-temperature application of car engine supercharging although such configuration for gas turbines has been also investigated [9-11]. The General Electric Company has obtained experimental data on a gas turbine engine enhanced by a RF wave rotor [12].

HOW DOES IT WORK INSIDE?

The wave process occurring inside the wave rotor channels is customarily illustrated by the wave diagram (space-time diagram), where the circular motion of the rotor channels is represented on paper by a straight translatory motion. It describes the rotor internal operation by tracing the trajectories of the waves and gas interfaces. The wave diagram is very useful for visualizing the wave process occurring inside the channels and also for determining wave rotor design parameters, i.e., port opening and closing times and their locations. The utility of the wave diagram is analogous to that of a velocity diagram for a conventional turbine or compressor.

Figure 5 represents NASA wave diagrams for through-flow (left) and reverse-flow (right) four-port wave rotors [13]. The figure shows the sequence of events occurring during one cycle within the channels moving in the upward direction. The journey of a channel of the wave rotor is periodic. The top of each wave diagram is considered to be looped around and joined to the bottom of the diagram. This requirement presents a fundamental challenge in the simulation and design of wave rotors. A successful prediction of the wave rotor implies that the state of the working fluid in the channel at the end of the cycle must be the same as that postulated at the beginning of the cycle.

To show how a four-port wave rotor works, the events occurring in one cycle of a TF four-port wave rotor are now described. The process begins in the bottom part of the left wave diagram in Fig. 5, where the channel is closed at both ends and contains low-temperature and low-pressure flow. For the RF configuration, the flow within the channel consists of a large part of the hot gas and a buffer layer separated by a contact surface. For the TF wave rotors, the gas fills the whole channel. As the channel gradually opens to the relatively low-pressure outlet port, an expansion fan originates from the leading edge of the outlet port and propagates into the channel, discharging only the gas to the turbine. The expansion fan reflects off the left wall and reduces the total pressure and temperature inside the channel further. This draws fresh air provided by the compressor into the channel when the inlet port opens. When the reflected expansion fan reaches the outlet port, it slows the outflow and reflects back as compression waves, while the outlet port then closes and halts the flow inside the channel. The compression waves form a single shock wave as they travel toward the inlet port. As the shock wave reaches the upper corner of the inlet port, it closes gradually. At this moment, the channel fluid is at rest relative to the rotor.

The above sequence of events is called the low-pressure part of the cycle (scavenging process). Its purpose is to deliver

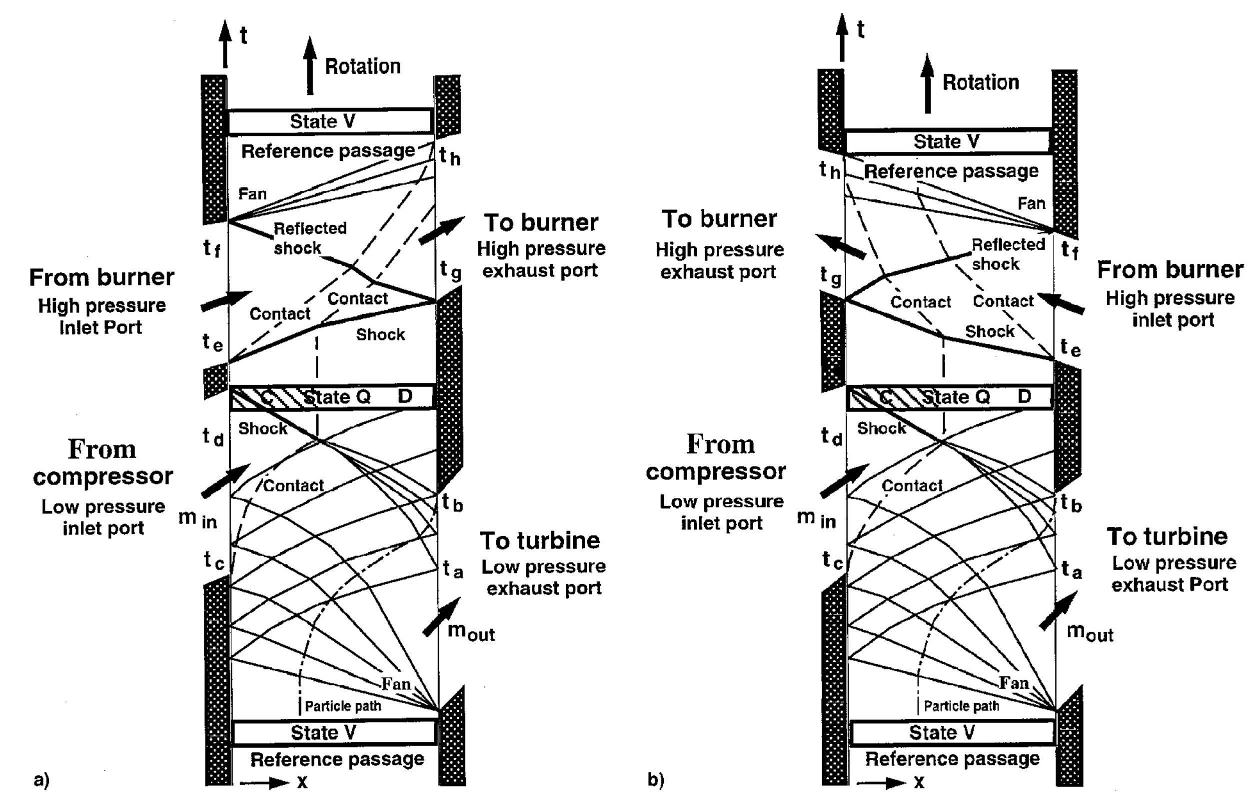

Figure 5: Wave diagrams for through-flow (left) and reverse-flow (right) four-port wave rotors, taken form Ref. [13]

a high-pressure gas into the turbine, partially purge the rotor channels, and ingest fresh air received from the compressor. In the high-pressure part of the cycle (charging process) that follows, the rotor channels are exposed to the burned gas arrived from the combustion chamber. This hot gas (driver) penetrates the channel when the inlet port is opened. Since the pressure of the burned gas is higher than the pressure in the channel (driven air), a shock wave is triggered starting from the lower corner of the inlet port. The shock wave runs through the channel and causes an abrupt rise of pressure inside the channel. As the shock wave reaches the end of the channel, the outlet port opens gradually and a reflected shock wave originates at lower outlet edge propagating back into the channel. The reflected shock wave compensates for the combustor pressure loss. The double-compressed flow behind the reflected shock wave leaves the wave rotor toward the combustion chamber. In the RF configuration the discharged flow into the burner is pure air, while in the TF configuration both air and once-burned gas are delivered into the burner. Detailed fluid flow investigations have suggested that approximately 30 to 50% of burned gas is recirculated to the combustion chamber in the TF configuration [14]. A favorite case is considered when the closure of the gas inlet port is timed with the arrival of the reflected shock wave. At this moment, an expansion fan originates from the upper corner of the inlet port and propagates toward the other end of the channel which eventually brings the channel flow to rest. When the expansion fan reaches the end of the channel, the outlet port gradually closes and the flow in the rotor channels stops. At this point, the flow with zero velocity is at nearly the peak pressure and temperature of the cycle. It is now ready to be discharged into the turbine by the low-pressure process.

HISTORICAL REVIEW

Recent advances and experiences obtained by the wave rotor community have renewed interest in this technology. These advances include new computational capabilities allowing accurate simulation of the flow field inside the wave rotor, and modern experimental measurements and diagnostic techniques. Improvements in aerodynamic design, sealing technologies, and thermal control methods have been sought. Recent developments in related unsteady flow and combustion processes of pulsed detonation engines have also provoked renewed interest. For this reason it is worthwhile to review the past and current work.

The Early Work (1906-1940)

The earliest pressure exchanger was proposed by Knauff in 1906 [15] in which he did not employ the action of pressure waves. The pressure exchanger introduced by him consisted of a cellular drum that rotates between two end plates containing several ports through which flows with different pressures enter and leave, exchanging their pressure. Knauff initially described rotor channels with curved blades and proposed inclined nozzles in the stator to achieve output shaft power (pressure exchange engine) besides pressure equalization inside the rotor. Reported by Pearson [16], Knauff in his second patent in 1906 [17] and Burghard in 1913 [18] proposed a simpler device in which pressure exchange takes place in long narrow channel configurations (pressure exchanger) known later as the Lebre machine following Lebre's patent in 1928 [19]. Today, the term static pressure exchanger is normally given to this type of device. Around 1928, Burghard proposed the utilization of pressure waves in another invention [20] that was termed the "dynamic pressure exchanger" to distinguish it from the static pressure exchanger. Here, the term "dynamic" implies the utilization of pressure waves in both compression and expansion processes taking place inside the rotor channels. However, difficulties mainly related to poor knowledge about unsteady flow processes limited the dissemination of the dynamic pressure exchanger concept [21] until World War II when Seippel in Switzerland implemented this concept into real engines, as discussed below.

The Comprex® Pressure Wave Supercharger (1940-Present)

Brown Boveri Company (BBC), later Asea Brown Boveri (ABB) and now Alston, in Switzerland has a long history in wave rotor technology. As reported by Meyer [22], their initial investigations in the beginning of the 1940s were aimed at implementing a wave rotor as a topping stage for a 1640 kW (2200 hp) locomotive gas turbine plant of British Railways [23-26]. They expected to obtain a power increase of 80% and a 25% efficiency increase based on the patents of Seippel [27-30]. This arrangement is shown in Fig. 6. The wave rotor had 30 channels rotating at 6000 rpm, with two opening ports on each side through which air and gas entered and left. It had originally shown a pressure ratio up to 3:1 and total efficiency of 69% in previous tests during 1941-1943, which could approximately result in a 83% efficiency for each compression and expansion process [22]. The first wave rotor worked satisfactorily, proving the concept of wave rotor machines. However, its performance when installed in the engine was far from expectations, mainly because of its inefficient design and crude integration [25].

Seippel's work also initiated the notion of using the wave

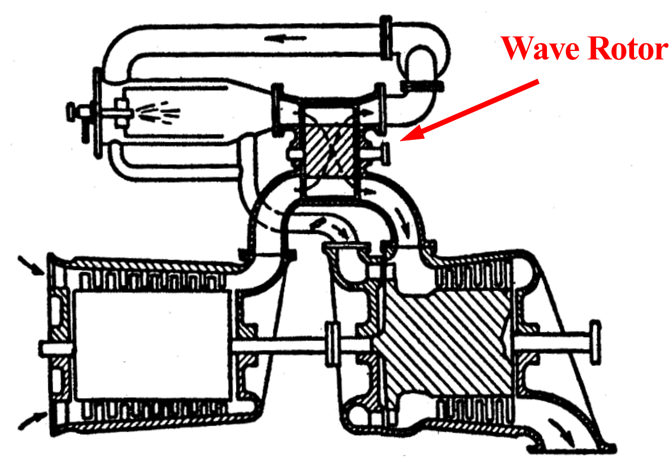

Figure 6: Wave rotor as a topping stage for the locomotive gas turbine, taken from Ref. [22]

rotor as a pressure wave supercharger for diesel engines. The extensive practical knowledge accumulated by BBC during investigations of gas turbine topping cycles was then used to develop pressure wave superchargers first by the ITE Circuit Breaker Company in the US [31-33]. In an effort jointly sponsored by the US Bureau of Aeronautics and ITE supervised of Kantrowitz of Cornell University and Berchtold of ITE, the first units were successfully manufactured and tested on vehicle diesel engines between 1947 and 1955. As a result of this success, a co-operative program with BBC was started in 1955 and BBC decided to concentrate on the development of pressure wave superchargers for diesel engines [34]. As a manufacturer of superchargers, BBC later continued the project in collaboration with the Swiss Federal Institute of Technology (ETH Zurich). While the first prototype was installed in a truck engine in 1971 [35], the supercharging of passenger car diesel engines was started in 1978 [36, 37] with a first successful test on an Opel 2.1 liter diesel engine [37, 38]. This supercharger was given the trade name Comprex® shown in Fig. 7. The port arrangement indicates the use of two operating cycles per revolution, shortening the rotor length and reducing thermal loads. The main advantage of the Comprex® compared to a conventional turbocharger is its rapid response to changes in engine operating conditions. Furthermore, as the efficiency of the Comprex® is independent of scale, its light weight and compact size make this device attractive for supercharging small engines (below about 75 kW or 100 hp) [39, 40]. By 1987, the first wide application of the Comprex® in passenger cars occurred in the Mazda 626 Capella [6, 41]. Since then, ABB's Comprex® pressure wave supercharger has been commercialized for several passenger car and heavy diesel engines. For instance, once Mazda produced 150,000 diesel passenger cars equipped with pressure wave superchargers [42]. The Comprex® has also tested successfully on vehicles such as Mercedes-Benz diesel car [7], Peugeot [34], and Ferrari [34].

The successful development of the Comprex® has been enabled by efforts of numerous researchers. Besides the above mentioned names, only some more are listed here: Gyarmathy [5], Burri [43], Wunsch [44], Croes [45], Summerauer [46], Kollbrunner [47], Jenny [48], Keller [49], Rebling [50], and Schneider [51]. Further references related to the development of the Comprex® by BBC [52-65] and other organizations [66-76] until 1990 can be found in the literature. By the end of the 1980s, when the Comprex® activity was transferred to the Mazda company in Japan [21, 77], researchers at ABB turned to the idea of utilizing wave rotor technology for gas turbine applications [78, 79].

During 1990s, a few groups continued the development of pressure wave superchargers. Nour Eldin and his associates at the University of Wuppertal Germany have developed a fast and accurate numerical method for predicting the unsteady-flow field in pressure wave machines, using the theory of characteristics [80-86]. Piechna et al. at the Warsaw University of Technology in Poland have developed experimentally validated one-dimensional and two-dimensional numerical codes to analyze the flow field inside the Comprex® [87-93].

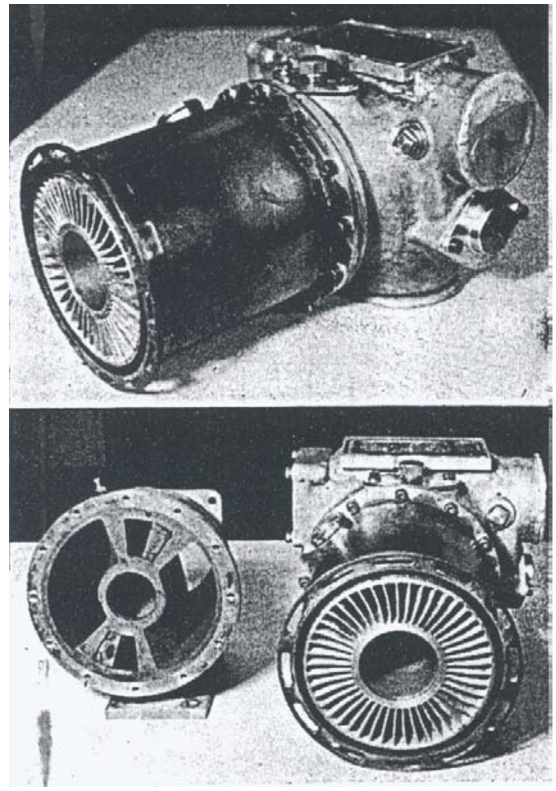

Figure 7: The Comprex®, taken from Ref. [39]

Piechna has also proposed a compilation of the pressure exchanger with the internal combustion wave rotor, presenting the idea of the autonomous pressure wave compressor [94]. Oguri et al. at Sophia University in Japan have performed measurements on a car gasoline engine supercharged by the pressure wave supercharger [95]. This effort sought to extend the application from diesel engines to gasoline engines and achieved a satisfactory increase of thermal efficiency of the supercharged engines. Guzzella et al. [40, 96-100] at ETH in Switzerland have developed a control-oriented model that describes the entire engine supercharged by pressure wave devices, with special emphasis on the modeling of transient exhaust gas recirculation phenomena. The experimentally validated model has introduced an optimized strategy to operate a supercharged engine with good drivability. Finally, an investigation of Comprex® supercharging on diesel emissions has been recently performed in Turkey [101], demonstrating that the Comprex® has the potential for reducing NO_X in diesel engines.

To date, the Comprex® has been the most commercialized of the wave rotor devices. The Comprex® development by BBC/ABB also has established fabrication techniques for wave rotors in commercial quantities and considered as a matured and reliable machine for internal combustion engine supercharging. For this application, BBC/ABB has solved difficult development challenges like sealing against leakages, noise, and the thermal stress problems. For instance, enclosing the rotor in a pressurized casing and using a rotor material with

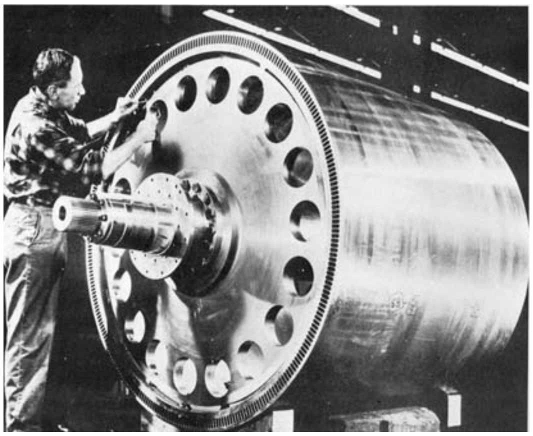

Figure 8: Photograph of the CAL Wave Supercharger, taken from Ref. [4]

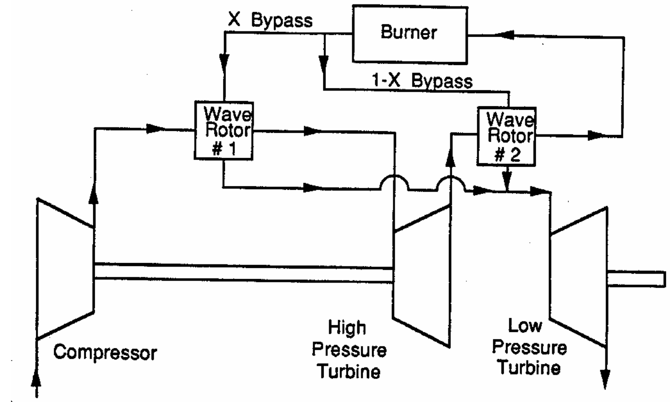

Figure 9: Schematic of a double wave rotor cycle, taken from Ref. [104]

a low thermal expansion coefficient over the operating temperature range has kept Comprex® leakages to an acceptable level [34]. Furthermore, several pockets have been cast into the end plates to control wave reflections and to achieve good off-design performance when engine speed changes [64].

In recent years, Swissauto WENKO AG in Switzerland has developed a modern version of the pressure wave supercharger [42]. This new generation of Comprex® known as the Hyprex® is designed for small gasoline engines. It benefits from new control features, enabling higher pressure ratios at low engine speeds, further reduced noise levels, and improvement of the compression efficiency at medium or high engine speeds. The Hyprex® has been successfully demonstrated in the SmILE (Small, Intelligent, Light and Efficient) vehicle which is a modified Renault Twingo, achieving very low specific fuel consumption and low emissions. ETH is collaborating closely in this effort by developing control systems for the proper operation of the device.

Cornell Aeronautical Laboratory and Cornell University (1948-2001)

Inspired by the cooperation with BBC in the late 1940s, work on unsteady-flow concepts was initiated at Cornell Aeronautical Laboratory (CAL). Among several novel concepts including development of energy exchangers for gas turbine cycles and various stationary power applications [102], the CAL Wave Superheater was built in 1958 and utilized until 1969 [26]. The 2 m diameter wave superheater used heated helium as the low molecular weight driver gas to provide a steady stream of high- temperature and high-pressure air for a hypersonic wind tunnel test facility. It compressed and heated air to more than 4000 K and up to 120 atm for run times as long as 15 seconds. Figure 8 is a photograph of this device. The CAL Wave Superheater was a landmark demonstration of the high temperature capabilities of wave rotor devices [26, 102].

Around 1985, Resler, a former member of the CAL Wave Superheater team, resumed the wave rotor research at Cornell University. His efforts and those of his group led to the development of new wave rotor concepts and analytic methods for three-port wave rotor diffusers [103], double wave rotor cycles [104], five-port wave rotors [104-111], and supersonic combustor aircraft engines using wave rotors [112]. Five-port wave rotors have shown significant potential for reducing NO_x in gas turbine engine applications. Figure 9 illustrates a double wave rotor in a gas turbine cycle. The idea of using a compound unit consisting of two (or multiple) wave rotors, one supercharging the other, is also reported in an early German patent by Müller in 1954 [113], as stated by Azoury [39].

Power Jets Ltd (1949-1967)

In parallel with but independent of Seippel's efforts in 1940s, Jendrassik, former chief engineer of the Gantz Diesel Engine Company of Budapest, was working on the development of wave rotor machines for gas turbine applications [25, 114-116]. He developed one of the first concepts for wave rotor applications to aircraft engines, proposing the wave rotor as a high pressure topping stage for early aircraft engines [117, 118]. His ideas stimulated the government-controlled company of Power Jets Ltd in the UK to become active in the wave rotor field in 1949. Even though the initial intent of Power Jets Ltd was to use wave rotor technology for IC engine supercharging, the interest was later extended to several other applications including air cycle refrigerators, gas turbines, pressure equalizers, and dividers [4, 21, 25, 116]. For instance, two prototype air-cycle refrigerators using wave rotors were designed and employed in gold mines in India and South Africa. They performed the same duty as equivalent vapor-cycle machines, but with lower weight and bulk. After Jendrassik's death in 1954, theoretical and experimental work continued at Imperial College, University of London, directed by Spalding and Barnes and also by Ricardo Company in the UK [25, 119]. The experimental divider test rig at Imperial College is shown in Fig. 10. Detailed information related to Power Jets Ltd efforts can be found in company reports listed in Ref. [4].

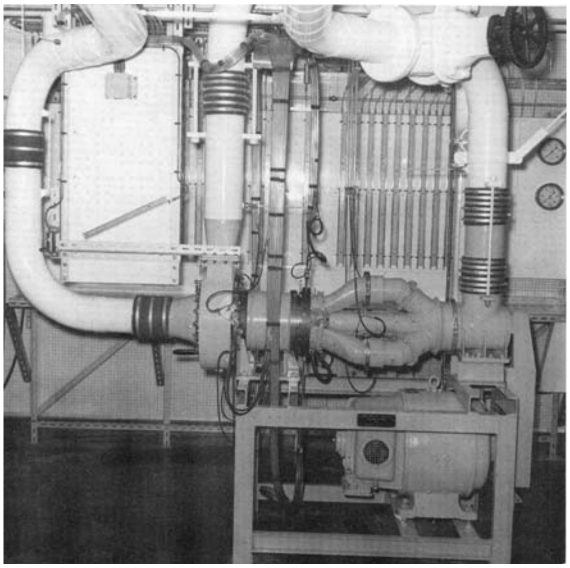

Figure 10: The experimental divider test rig at Imperial College, taken from Ref. [4]

Computational methods and facilities were too little developed at the time, and theoretical methods yielded little progress. Tedious manual cycle analyses were almost impractical. Spearheading the development of CFD methods, Spalding of Imperial College formulated a numerical procedure for wave rotors considering the effects of heat transfer and friction. It utilized novel features to ensure solutions free from instabilities and physical improbabilities [25]. Based on this numerical model, a computer program was developed by Jonsson [120] and it was successfully applied to pressure exchangers [121-123]. Spalding's students, Azoury [124] and Kentfield [125], continued their efforts on different theoretical aspects of pressure exchangers [4, 21, 25, 39, 116, 123, 126-129] despite the dissolution of Power Jets Ltd in 1967 [25].

Ruston-Hornsby Turbine Company: The Pearson Rotor (Mid 1950s - 1960)

In the U.K. of the mid 1950s, besides the work at Power Jets Ltd and Imperial College, the Ruston-Hornsby Turbine Company, manufacturer of diesel engines and industrial gas turbines, supported the construction and testing of a different kind of wave rotor designed by Pearson [130, 131]. This unique wave rotor, known as the wave turbine engine or simply the wave engine, has helical channels that change the direction of the gas flows producing shaft work similar to a conventional turbine blade. Pearson designed and tested his wave rotor in less than a year, shown in Fig. 11. The rotor has a 23 cm (9") diameter and a 7.6 cm (3") length. The engine worked successfully for several hundred hours in a wide range of operating conditions (e.g., 3000-18000 rpm) without variable porting, and produced up to 26 kW (35 hp) at its design point with a cycle peak temperature of 1070 K and a thermal efficiency of around 10%. While the performance results were slightly less than the expected design performance (mainly due to the combined effect of excessive leakage and incomplete scavenging), higher performance seemed to be possible with more careful design and development. The design of the engine was based on many wave diagrams using the method of characteristics that accounted for all internal wave reflections. The engine utilized extra ports and injection nozzles to control and cancel unwanted reflected waves. The engine had a length of only one third of its diameter despite having only one cycle per revolution [16]. The sealing and bearings were carefully adapted considering rotor thermal expansion. Unfortunately, the engine was accidentally wrecked due to over speeding from an improperly connected fuel line. Tragically, the wave rotor project was canceled when success seemed so close, but the company had financial difficulties. Despite the technical success achieved, Pearson failed to attract additional funding for this radical idea.

In the early history of wave rotor technology, the Pearson rotor and the Comprex® have worked efficiently over a wide range of operating conditions [25, 34, 115], demonstrating good off-design performance, while the Wave Superheater was

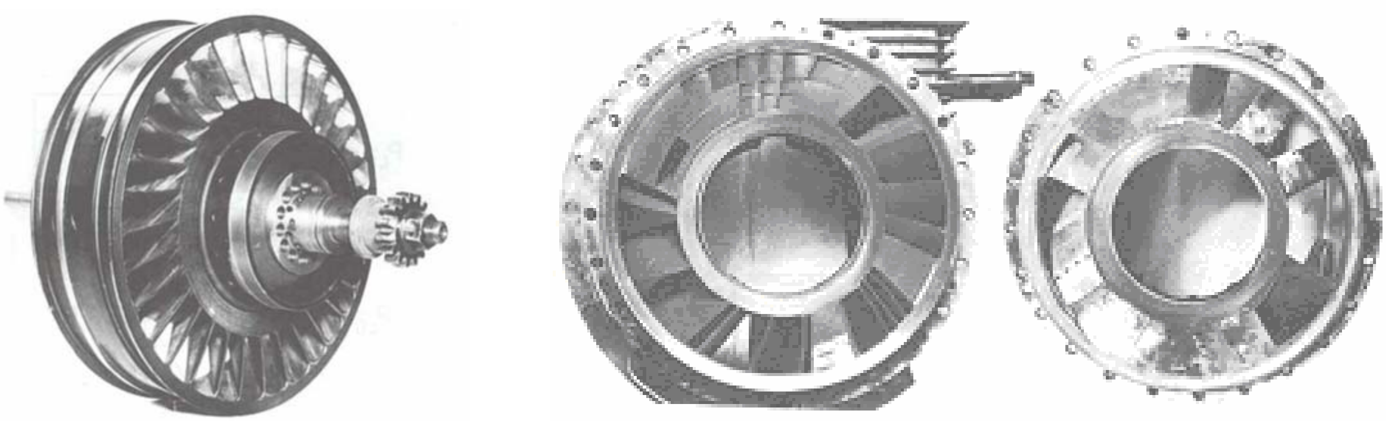

Figure 11: The Pearson rotor (left) and rear and front stator plates (right), taken from Ref. [130]

an equal success. Nevertheless, the Pearson is a notable wave rotor for producing a significant power output in addition to being a successful pressure exchanger.

General Electric Company (1956-1963)

While Pearson was developing hisl wave engine, General Electric Company (GE) in the US initiated a wave rotor program in 1956 [12]. The work was motivated by earlier work at NASA Langley initiated by Kantrowitz and continued by Huber [132] during the development of a wave engine in the early 1950s and later in 1954-1956 developing pressure gain combustors (constant volume combustion) [12]. GE studied a new configuration of wave rotor in which combustion took place inside the rotor channels (internal combustion wave rotors). Such an arrangement eliminates the external combustion chamber used in the gas turbine cycle, resulting in a significantly lower weight, less ducting, and a compact size. In the period of 1956 to 1959, the methods used at NASA were analyzed, improved and applied to the design and fabrication of the first internal combustion wave rotor demonstrator. As reported by Weber [2], the test rig was first tested at the California Advanced Propulsion Systems Operation (CAPSO) of GE. After 20 seconds of operation, the rotor seized between the end plates causing an abrupt stop. The test demonstrated the difficulty of clearance control between the end plates and rotor during thermal expansion. Rotor expansion is an especially challenging problem in the design of wave rotors. While the running clearance between the end plates and rotor must be kept as small as possible, the rotor tends to expand thermally due to hot gases in the rotor. Henceforth, GE resorted to inferior rubbing seals, and tested only pressure-exchange configurations from 1960 to 1961 [12]. Despite flow leakage, respectable wave rotor overall pressure ratios of 1.2 to 1.3 were achieved. Meanwhile, a feasibility study was initiated for reducing compressor stages of a T-58 GE-06 engine by using a wave rotor. It showed a considerable reduction in overall engine weight and cost, and a 15% reduction in specific fuel consumption rate. These results motivated a conceptual design layout of such an advanced engine. However, further rig experiments revealed other mechanical and aerodynamic shortcomings including start-up, bearing durability, fuel system complications and control [8].

GE also pursued a shaft work output wave rotor. Over the period from 1961 to 1963, Klapproth and his associates at GE in Ohio fabricated and tested a wave engine using air-gap seals. An ideal wave diagram of this engine is shown in Fig. 12. The engine worked continuously, but it did not produce the anticipated net output power. It is believed that insufficient attention was given to account for internal wave reflections, thus, the flow field calculations were inaccurate [34]. Simplifications were unavoidable at that time and generation of wave diagrams by hand required considerable time and effort and small design changes necessitated a lengthy recalculation. Although the Klapproth rotor did not produce the expected performance, it clearly demonstrated the possibility of the complete exchange of energy within the wave rotor. GE development of the wave rotor was canceled in 1963 due to

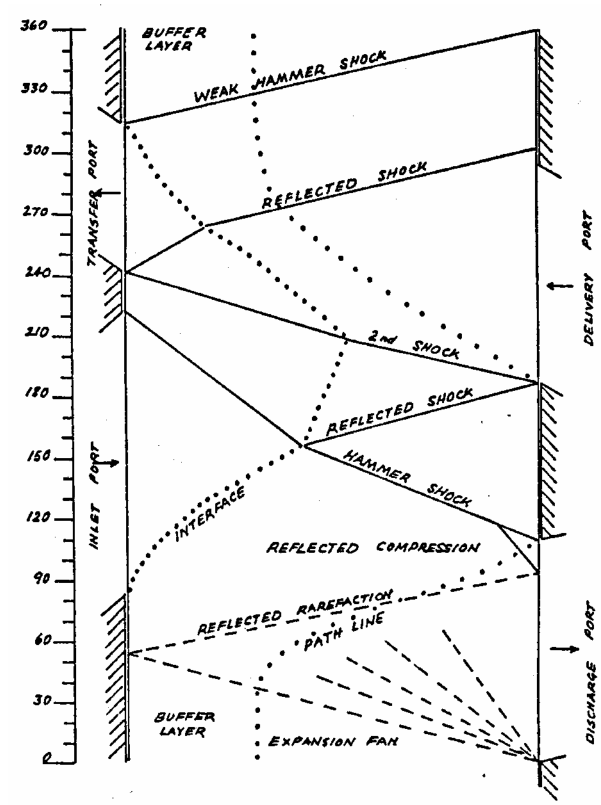

Figure 12: Ideal wave diagram of the Klapproth rotor, taken from Ref. [12]

shifting funds from turbine engine development to space exploration and rocket propulsion [2], and GE's commitment to pursue large engine development exclusively [12, 132].

General Power Corporation (Mid 1960s - 1984)

In the mid 1960s, General Power Corporation (GPC) started a wave rotor program originally intended for a road vehicle engine application [34]. Over a period of about 20 years, GPC spent considerable time and money to successfully design and develop wave rotors. The work was initially supported by Ford Motor Company and later by the Department of Energy (DOE) and the US Defense Advanced Research Program Projects Agency (DARPA). Unfortunately, the GPC work is poorly documented [133]. As stated by Taussig [34, 115], while the GPC rotor shared some of the features of the Klapproth and Pearson rotors, it differed in several aspects. Figure 13 illustrates an ideal wave diagram of the GPC rotor, intended to produce reactive shaft power utilizing curved blades. Its performance suffered from excessive blade curvatures, lack of control of reflected waves within the device, and the absence of any strong impulsive loading of the rotor from inlet manifolds to produce shaft work. The latter was in contrast with the Pearson rotor that relied heavily on impulsive loading of the rotor blades to achieve power output. Furthermore, the GPC rotor had inadequate control on maintaining high off-design performance. Although

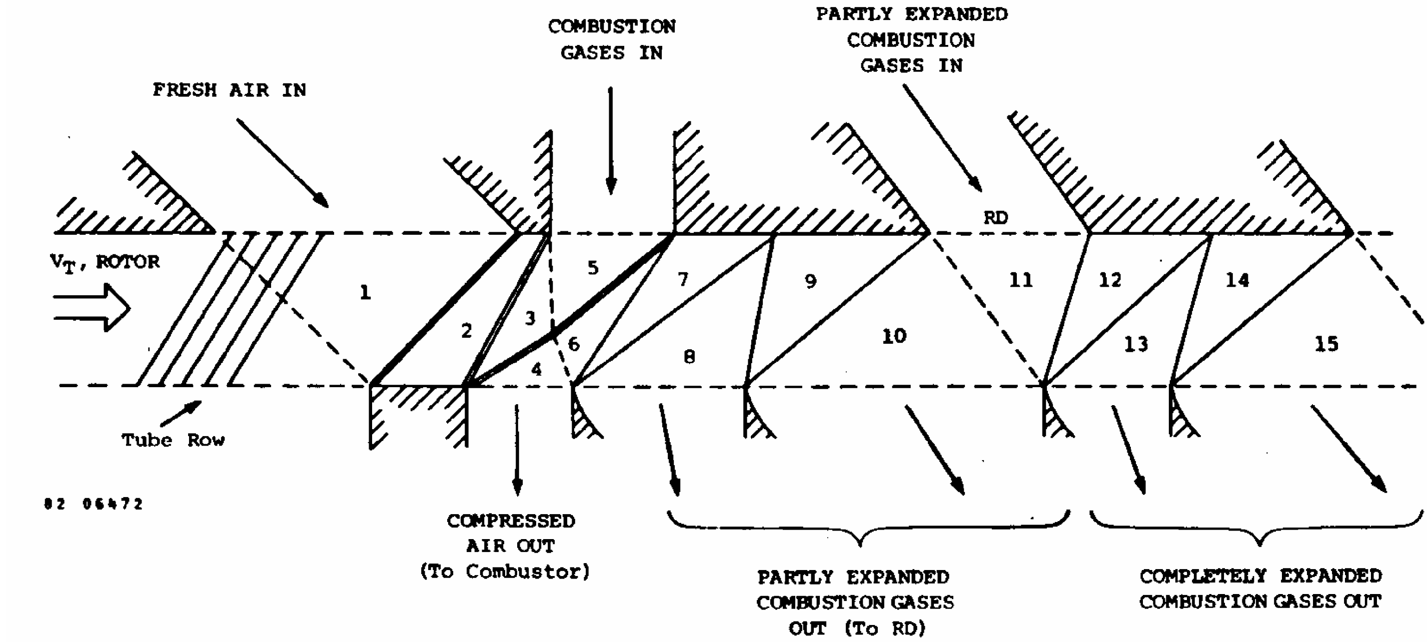

Figure 13: Ideal wave diagram of the GPC rotor, taken from Ref. [115]

GPC developed a computer code to obviate manual wave pattern design, accurate calculations were still tedious. Ultimately, Ford withdrew its support from the wave rotor research [134] and GPC discontinued development of the wave engine in the early 1980s.

Rolls-Royce (1965-1972)

In the mid 1960s, Rolls-Royce (RR) in the UK began numerical and experimental wave rotor research [34]. BBC cooperated with RR in the development of pressure exchange wave rotors as topping spools in gas turbine applications [9], with Berchtold of the ETH and Spalding of Imperial College serving as consultants [21]. Considerable efforts were made to design a wave rotor as a topping stage for a small helicopter engine (Allison Model 250) [135]. The BBC-RR engine utilized a reverse-flow wave rotor incorporated into a single turbine cycle. This was somewhat different from the cycle suggested by Berchtold and Lutz [68] in BBC gas-turbine-topping investigations ,which employed a through-flow wave rotor integrated with both low-pressure and high-pressure turbines. BBC's interests in wave rotors at that time were mostly related to development of small gas turbines for passenger cars, beset by poor efficiencies at sizes of 100 kW and smaller [26]. Similar to previous wave rotor efforts, rotor designs protracted manual design methods. While the enhanced engine operated nearly as predicated, performance suffered from leakage [34]. Other difficulties related to the start-up and control are reported [8]. The program was abruptly canceled in 1972 amidst severe company financial difficulties [9]. As stated by Kentfield [21], contemporaneous rapid progress in turbomachinery technology may have disfavored high-risk projects, both at RR and GE.

Mathematical Science Northwest Inc. (1978-1985)

In the late 1970s Mathematical Science Northwest Inc. (MSNW, later Spectra Technology Inc., and now STI Optronics Inc.) investigated various applications of wave rotors [26]. Under the sponsorship of DOE and DARPA, they considered a broad range of stationary power systems such as magnetohydrodynamic cycles (MHD) [34], combined cycles integrated with gasification plants [136], pressurized fluidized bed (PFB) power systems [137], and also propulsion and transportation applications [115]. Significant numerical and experimental efforts included developing a laboratory wave rotor [138-141], shown in Fig. 14. With diameter of 45 cm, it consists of 100 channels each with a 40 cm length. It is a four-port wave rotor with two additional small ports provided to cancel pressure waves at critical rotor locations providing more uniform port flows and a higher transfer efficiency [141]. A pressure ratio of approximately 2.5 was achieved. Besides successful tests using several configurations (clearance variations, port sizes, etc.) and various operating conditions, experiments were designed to verify the scaling laws for predicting the performance of larger machines [136]. The MSNW wave rotor was initially designed based on the method of characteristics, but later a one-dimensional unsteady computer code (the FLOW code) was used for optimizations and modifications of the MSNW design [115]. The modifications led to improvement in obtaining a very good agreement between the numerical and experimental results in a wide range of operating conditions. The FLOW code which was developed specifically for both pure pressure exchanger wave rotor and wave engine analyses, uses the flux-corrected transport algorithm solving Euler equations accounting for heat

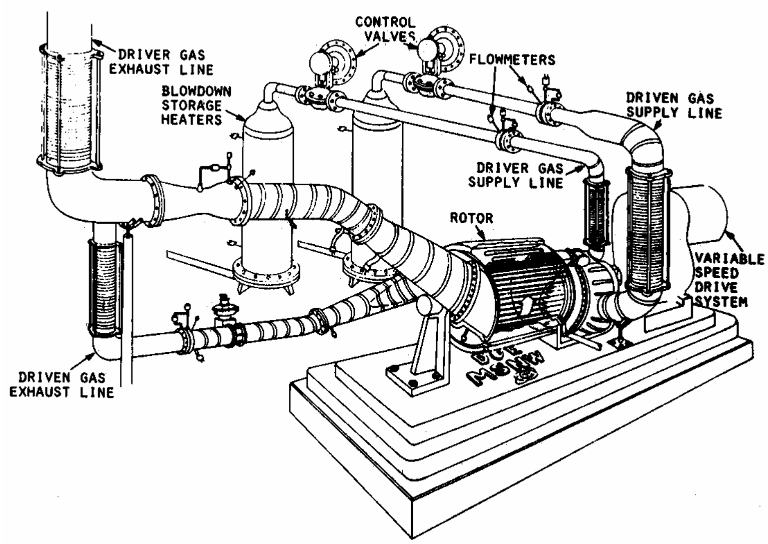

Figure 14: Schematic of the MSNW wave rotor experimental set up, taken from Ref. [26]

transfer, viscosity, gradual port opening, and flow leakage. The sensitivity of wave rotor performance to tip speed, port placement and size, inlet and outlet flow conditions, channel geometry, number of channels, leakage, and heat transfer was analyzed for both on-design and off-design conditions. For instance, it was concluded that heat transfer losses were negligible and leakage was recognized as a key problem for efficient wave rotor operation. Numerical work has been also reported for a nine-port wave rotor concept to resolve the problem of nonuniform port flows and poor scavenging.

MSNW also produced preliminary wave rotor designs for a small turbofan engine generating 600 lb thrust at sea level condition [115, 142], illustrated in Fig. 15. Performance calculations for both on-design and off-design flight conditions using a cycle performance code and the FLOW code simulation have predicted significant performance improvements of such an enhanced engine. No new material development for such combined engines was required.

The wave rotor activity at MSNW was discontinued in the mid 1980s. No specific reasons for this cancellation are reported.

Naval Postgraduate School (1981-1986)

In 1981, the Office of Naval Research (ONR) agreed to monitor a joint DARPA/ONR program to evaluate the wave rotor concept and its potential application in propulsion systems [132]. Following this decision, Turbopropulsion Laboratory (TPL) at Naval Postgraduate School (NPS), directed by

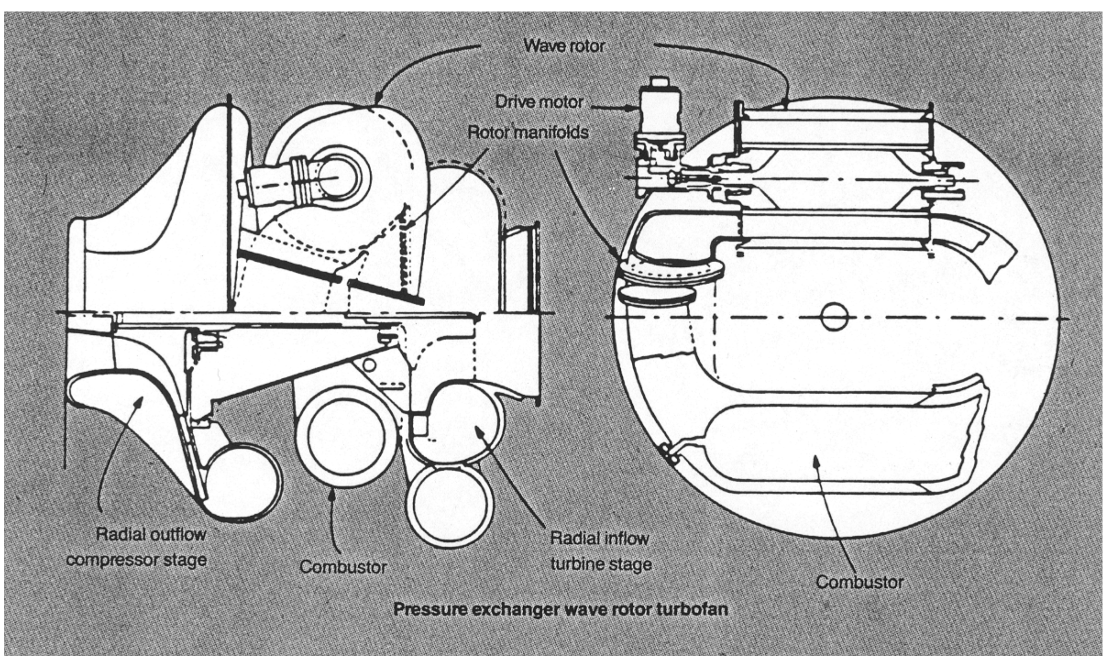

Figure 15: Conceptual design of a turbofan engine incorporating a wave rotor, taken from Ref. [142]

Shreeve, started an extensive numerical and analytical wave rotor program. To support the accuracy of the computational results, the wave rotor apparatus formerly used by Klapproth at GE was transferred to TPL and some preliminary tests were carried out out. It is reported that the rotor produced some shaft work running at approximately 5000 to 6000 rpm [143]. No further experimental details are reported.

For numerical simulations, two different approaches to the solution of the unsteady Euler equations were examined in the overall program. First, Eidelman developed a two-dimensional code based on the Godunov Method to analyze the flow in wave rotor channels [144-147]. Unlike contemporary one-dimensional approaches [148], the two-dimensional code showed the effect of gradual opening of the channels. The main conclusion of these studies is that if the channels are straight, the flow remains nearly one-dimensional, which in turn leads to minimal mixing losses caused by rotational flow in the channels [149]. However, when the channel of the wave rotor is curved, even an instantaneous opening of the channel does not lead to the development of a one-dimensional flow pattern with small losses. For faster computations, a one-dimensional, first order time-accurate code was introduced by Mathur based on the Random Choice Method for solving the Euler equations [150, 151]. The unconditionally stable code, called WRCOMP (wave rotor component), calculated the unsteady-flow process inside the wave rotor, inlet and outlet opening times and other useful design parameters required for a preliminary design. The outputs from WRCOMP are used in a second program, called ENGINE, for turbofan jet engine performance calculations [152-154]. The results confirmed the significant performance improvement expected by integrating a wave rotor into a turbofan engine. Some improvements to WRCOMP code were later begun [155, 156], but the wave rotor research was terminated around 1986. NPS also sponsored the most comprehensive wave rotor conference in 1985 [157], which reviewed much of the history to that point.

NASA Glenn Research Center (1988-Present)

Since the late 1980s, a sustained research program at NASA Lewis (now Glenn) Research Center (GRC), collaborating with the US Army Research Laboratory (ARL), Rolls-Royce Allison has aimed to develop and demonstrate the benefits of wave rotor technology for future aircraft propulsion systems [8].

In 1993, using a thermodynamic approach to calculate the thermal efficiency and specific power, Wilson and Paxson [158] published a feasibility study for topping jet engines with wave rotors. Applied to the case of an aircraft flying at Mach 0.8 , they have shown that a wave-rotor-topped engine may gain 1…2% in efficiency and 10…16% in specific power compared to a simple jet engine with the same overall pressure ratio and turbine inlet temperature. Additionally, Paxson developed a quasi-one-dimensional gasdynamic model and a computational code to calculate design geometry and off-design wave rotor performance [159, 160]. The code uses an explicit, second order, Lax-Wendroff type TVD scheme based on the method of Roe to solve the unsteady flow field in an axial passage for time-varying inlet and outlet port conditions. It employs simplified models to account for losses due to gradual passage opening and closing, viscous and heat transfer effects, leakage, flow incidence mismatch, and non-uniform port flow field mixing. In order to verify wave rotor flow predictions and to assess the effects of various loss mechanisms [161, 162], a three-port wave-divider machine was constructed and tested [163-165] in a new wave rotor laboratory facility at GRC. Concurrently, the non-ideal behavior and losses due to multi-dimensional effects were studied by Welch [166-168] and Larosiliere [169-171]. Welch has also established macroscopic and passage-averaged models to estimate the performance enhancements of wave rotors [13, 172]. Based on experimental data, Paxson further improved the one-dimensional model [161, 162, 173, 174] and used it to evaluate dynamic behavior, startup transients, and channel area variation [175-178]. This model was then used as a preliminary design tool to evaluate and optimize a four-port wave rotor cycle for gas turbine topping [179]. This through-flow cycle was chosen based on several perceived merits, including relatively uniform rotor temperature, and the feasibility of integration with gas turbomachinery. As a result of these studies, a new four-port wave rotor was designed and built [180] to test the performance of this concept under scaled laboratory conditions. A photograph of NASA four-port wave rotor is shown in Fig. 16. However, a study by Rolls-Royce Allison discussed below indicated that thermal loads on the rotor and ducting predicted for the NASA wave rotor cycle in real engine conditions may be difficult to manage. In response, Nalim and Paxson [181, 182] proposed an alterative cycle with a combustor bypass significantly lowering thermal loads.

Additional studies of the performance benefits of wave rotor topped gas turbines have been reported. In 1995, Welch et al. [14] predicted a 19…21% increase in specific power and a 16...17% decrease in specific fuel consumption compared with the baseline engines in performance calculations for small (300 to 500 kW) and intermediate (2000 to 3000 kW) wave-rotor-enhanced turboshaft engines. The same calculations for a wave-rotor-enhanced large turbofan engine, equal in thrust to the

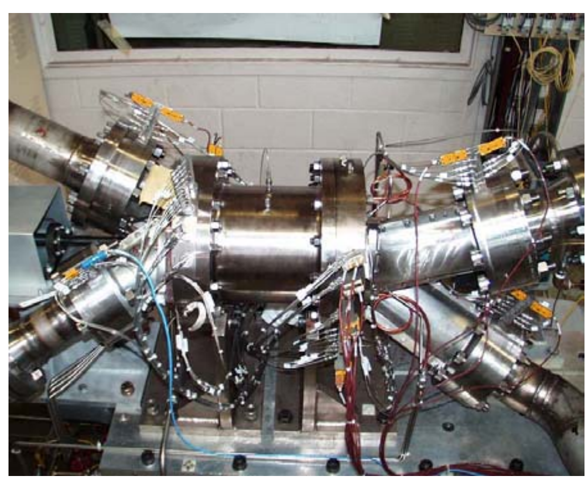

Figure 16: Four-port wave rotor of NASA

baseline engine, have shown a 6…7% reduction in thrust specific fuel consumption. Welch has also studied the possibility of curving the channels to create a wave turbine [183, 184].

In 1995, Nalim at NASA published a feasibility assessment of combustion in the channels of a wave rotor, for use as a pressure-gain combustor [185]. Combustion prediction capability was added to the wave rotor code by Nalim and Paxson [186], enabling the exploration of wave cycles involving both detonation and deflagration modes of combustion. For uniform mixtures, a single reaction progress variable is utilized. Multiple species are represented for a variable fuel-air ratio in deflagration modes. Mixing controlled reaction is combined with a simple eddy diffusivity model. Other notable features that were incorporated are temperature kinetics factors and a simple total-energy based flammability limit [187]. The performance of detonative and deflagrative cycles was studied by combined CFD and system simulation. It was determined that deflagrative combustion with longitudinal fuel stratification could be accomplished over a reasonable time in wave rotors.

The current NASA wave rotor research has been mostly focused on experimental tests with special attention to sealing technology [188-190], identified as a critical challenge in high-pressure wave rotor design.

Rolls-Royce Allison (1990-Present)

In 1996, Snyder and Fish [10, 191] of Allison Engine Company evaluated the Allison 250 turboshaft engine as a potential platform for a wave rotor demonstration, predicting an 18...20% increase in specific power and a 15...22% decrease in specific fuel consumption. They used a detailed map of the wave rotor cycle performance accomplished by Wilson and Paxson [8, 158, 179]. Allison (by now Rolls-Royce Allison) has also studied transition duct designs for integration with turbomachiney [192, 193]. This was later followed by investigations of pulse detonation wave rotors in the newly formed Allison Advanced Development Company (AADC). A novel four-port device is proposed [194] for supersonic turbofan engines [195], and was investigated in collaboration with Indiana University Purdue University Indianapolis (IUPUI) as discussed below.

University of Florida (1992-1998)

Motivated by NASA wave rotor successes, Lear at the University of Florida initiated analytical and numerical methods to investigate different configurations of wave rotors. His team developed an unsteady two-dimensional numerical code using a direct boundary value method for the Euler equations to analyze the flow in wave rotors and their adjoining ducts, treating the straight or curved channel walls as constraints imposed via a body force term [196]. The code was later used to simulate the flow field of the three-port NASA wave rotor. They also introduced a preliminary design method for selecting the wave engine inflow and outflow blade angles. Furthermore, an analytical thermodynamic description of wave rotors was developed [197], which predicted potential increase in specific power of 69% and a 6.8% increase in thermal efficiency over a conventional gas turbine topped by a wave engine. A parametric study of gradual opening effects on wave rotor compression processes is reported, too [198].

ONERA in France (1995-1999)

Fatsis and Ribaud at the French National Aerospace Research Establishment (ONERA) have investigated wave rotor enhancement of gas turbines in auxiliary power units, turboshaft, turbojet, turbofan engines [11, 199], accounting for compression and expansion efficiency, as well as mixing and pressure losses in the ducting. Their results show the largest gains and efficiency for engines with a low compressor pressure ratio and high turbine inlet temperature, such as turboshaft engines and auxiliary power units. These results are consistent with those obtained by NASA GRC [200]. They hve also developed a one-dimensional numerical code based on an approximate Rieman solver taking into account viscous, thermal, and leakage losses [11, 201], and applied it to three-port, through-flow, and reverse-flow configurations.

RECENT ACADEMIC WORK

Besides ongoing research mainly at NASA, AADC, and ETH Zurich, a few universities have been conducting wave rotor research. To the knowledge of the authors, the universities listed below are active in this field.

Purdue School of Engineering and Technology, IUPUI (1997-Present)

Recent research at Indiana University Purdue University Indianapolis (IUPUI) by Nalim and coworkers has focused on internal combustion wave rotors, following initial work at NASA described before. Deflagrative combustion with longitudinal fuel stratification has yielded a wave rotor geometry competitive with pressure-exchanger designs using a separate combustor [187, 202]. Nalim has highlighted the importance of leakage flows and thermal management of end-wall temperatures illustrating the impact of the hot ignition gas and the cold buffer zones on the end walls. This is consistent with the major challenges revealed by the ABB experiment [79]. Radial stratification [203] using a pre-combustion partition has been proposed to introduce a relatively cooler buffer zone close to the leakage gaps, reducing hot gas or fuel leakage to the rotor cavity. Figure 17 is a contour plot of the temperature contour from a simulation of deflagrative combustion in a stoichiometric partition region propagating into a leaner mixture in the main chamber. Above and below the partitions, there is no fuel, and gas may leak out or in without danger of overheating or pre-ingnition. These thermal management approaches are possible utilizing extensive cycle design studies and analysis, and seek to alleviate the challenges

Figure 17: Temperature distribution of partition exit flow, taken from Ref. [204]

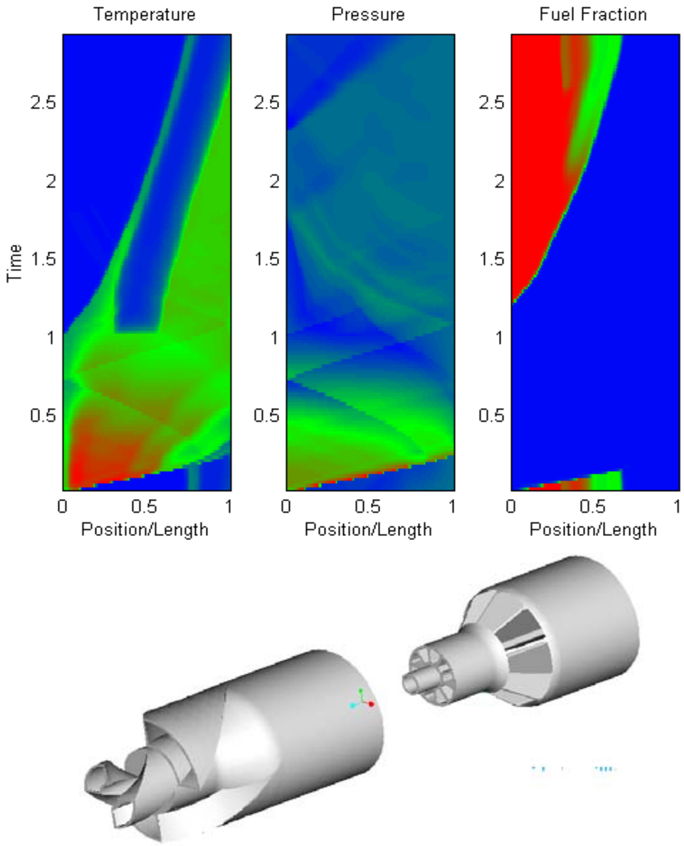

Figure 18: Rotary Wave Ejector Pulse Detonation Engine, taken from Ref. [210]

previously recognized by ABB and NASA. This technique also helps burn leaner mixtures, resulting in reduced NO_X emissions, similar to other pilot combustion or lean-burn techniques in conventional engines [204]. For this approach, radial leakage flows [205] and different combustion models [206] have been studied in detail. These ideas have not yet been tested experimentally.

Detonative combustion cycles for propulsion engines have been also studied [207, 208]. Interest in detonative combustion initially focused on pulsed detonation engines (PDE) has evolved to the consideration of the wave rotor as an effective implementation of the concept [209], and a means of overcoming challenges to PDE concepts that involved integration with conventional turbomachinery. In effect the wave rotor provides automatic high-speed valving, nearly steady inflow and outflow, and the use of one or few steady ignition devices for multiple tubes. However, detonative combustion is fundamentally restricted to highly energetic mixtures and sufficiently large passage widths, and generates strong pressure waves. This results in the outflow being highly non-uniform in pressure, velocity, and possibly temperature. To better utilize the output of a wave rotor PDE, it has been proposed to add an ejector element to the wave rotor [210]. The rotary wave ejector admits bypass air after the detonation tubes to transfer energy and momentum. Numerical simulations using a quasi-one-dimensional code, modified to account for radial-type bypass flows, have shown that the specific impulse at static thrust conditions can be doubled, after accounting for flow-turning and shock losses, comparing with an equivalently loss-free PDE cycle. A sample wave diagram and a schematic sketch are given in Fig 18, where the cold ejector gas flow is clearly distinguishable.

IUPUI has also investigated [211] the four-port detonation wave rotor proposed by AADC [194], in which a recirculation duct allows air that is compressed by the shock of a detonation wave to be reinjected with fuel. Air-buffer regions both between the fuel/air-combusted gas interface and at the exit end plate are inherent in the cycle design, allowing self-cooling of the walls. The inflow and outflow of this engine concept is designed to be nearly uniform and acceptable to modern turbines, compared to conventional rotary detonation cycles, as shown in Fig. 19.

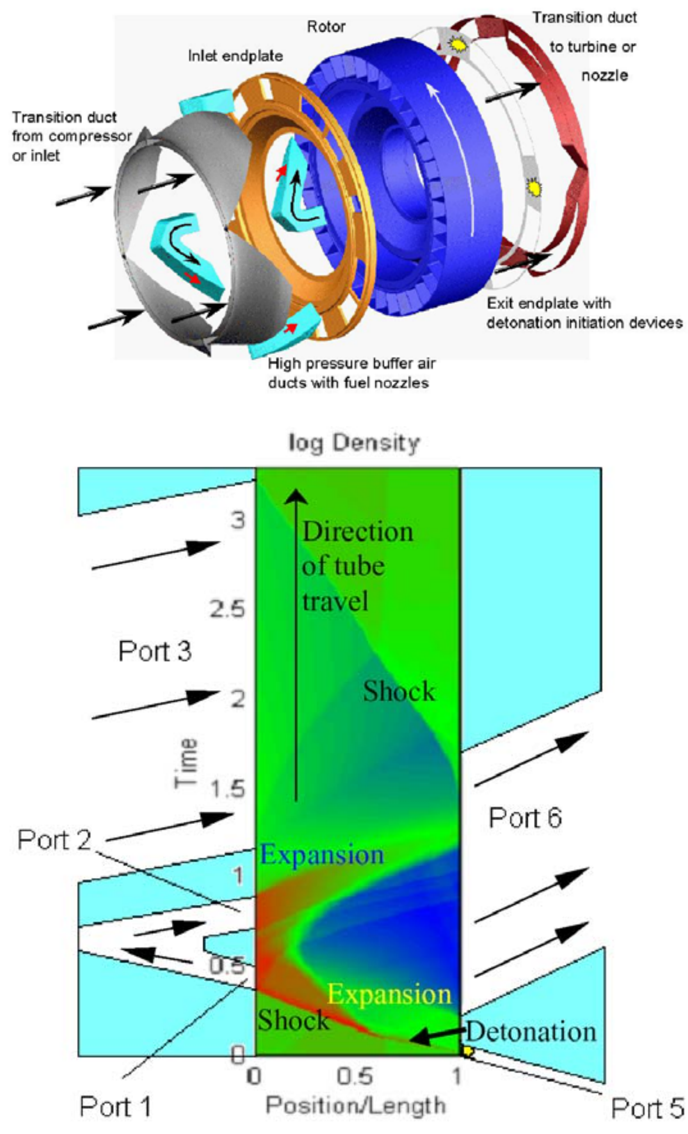

Figure 19: Wave Rotor Pulse Detonation Engine, the 'CVC' Engine, taken from Ref. [211]

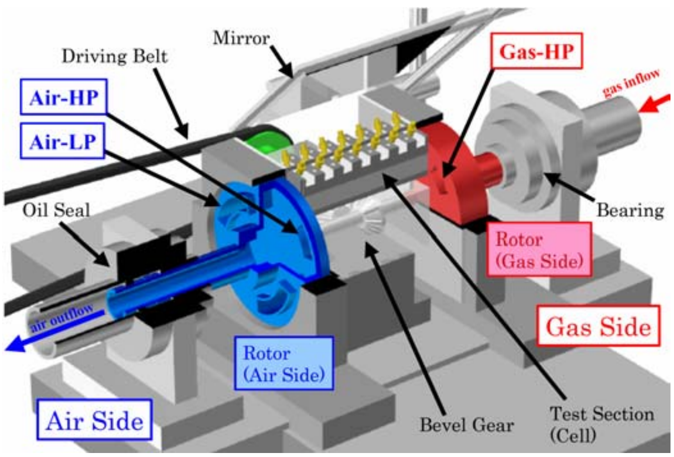

Figure 20: University of Tokyo single-channel test rig, taken from Ref. [214]

A computational and experimental program is currently being conducted at IUPUI in collaboration with AADC to investigate the combustion process and performance of a wave rotor with detonative and near-detonative internal combustion [212]. A preliminary design method based on a sequence of computational models has been developed to design wave processes for testing in an experimental test rig.

University of Tokyo (2000-Present)

Nagashima et al. have developed one-dimensional [213] and two-dimensional [214] numerical codes to simulate the flow fields inside through-flow four-port wave rotors, including the effects of passage-to-passage leakage. The codes have been validated with experimental data obtained by a single-channel wave rotor experiment. The test rig, shown in Fig. 20, consists of a stationary single tube, and two rotating plates connected to a shaft driven by an electric motor. This group has also explored the idea of using wave rotors for ultra-micro gas turbines manufactured using microfabrication technology [215].

Michigan State University (2002-Present)

The MSU wave rotor group has initiated studies to evaluate wave rotor technology benefits for several thermal cycle applications. Two unrecuperated microturbines (30 and 60 kW) implementing various wave-rotor-topping cycles were predicted to have overall thermal efficiency and specific work enhancement up to 34% for the smaller engine and 25% for the larger engine, using a four-port wave rotor with a compression ratio of 1.8 [216]. Similar approach has predicted an improvement up to 15% of overall efficiency and specific thrust in a turbojet engine using the wave-rotor-topping cycle of 30 kW microturbine flying at an altitude of 10,000m at Mach 0.8 [217]. Using predicted performance results, the team has also developed an analytical preliminary design procedure for the critical high-pressure phase of four-port wave rotors [218, 219].

Recently, a unique and cutting-edge application of wave rotors in refrigeration cycles using water (R718) as a refrigerant

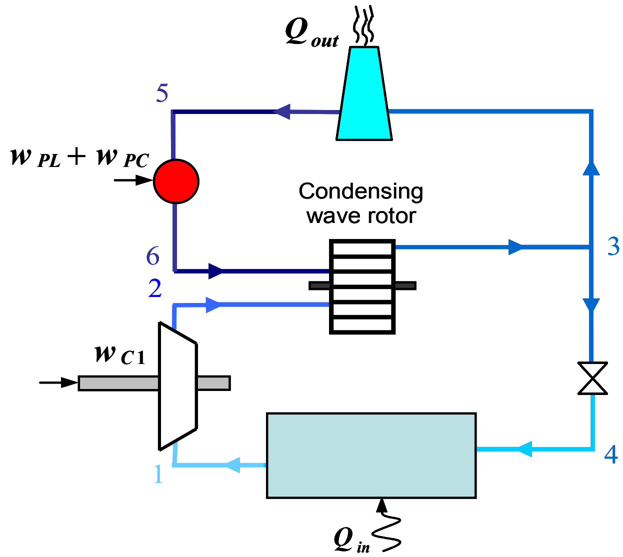

Figure 21: Schematic of a R718 cycle enhanced by a three-port condensing wave rotor substituting for the condenser and one compressor stage, taken from Ref. [220]

has been studied [220-223]. In fact, the wave-rotor implementation can increase efficiency and reduce the size and cost of R718 units. A three-port wave rotor has been introduced as a condensing wave rotor that employs pressurized water to pressurize, desuperheat, and condense the refrigerant vapor - all in one dynamic process. In addition to the possibility of an additional rise of the vapor pressure, the condensing wave rotor eliminates the need of a bulky condenser because full condensation occurs inside the rotor channels. Furthermore, adding a condensing wave rotor to a water refrigeration cycle allows for a lower pressure ratio of the compressor, which is crucial for the R718 chiller technology. Figure 21 shows a schematic of a R718 cycle using a three -port condensing wave rotor.

Investigations of the feasibility and potential of integrating four-port wave rotors in microfabricated gas turbines are also being pursued at MSU [224]. Ultra-micro gas turbines (UμGT) have shown difficulties in obtaining high overall thermal efficiency and output power, resulting from miniaturization. Utilizing wave rotor technology to improve the performance of UμGT as an appropriate solution is suggested. The wave rotor can enhance both the overall thermal efficiency and specific work output, if the wave rotor compression efficiency is higher than that of the baseline engine compressor. MSU studies have shown that an efficiency of the compression between 70% and 80% can be achieved in an ultra-micro wave rotor. Several different advantageous conceptual designs for a four-port wave rotor integrated into a baseline UμGT are studied. For instance, Fig. 22 shows a wave rotor added at outer diameter of the disc of the well-known MIT microengine.

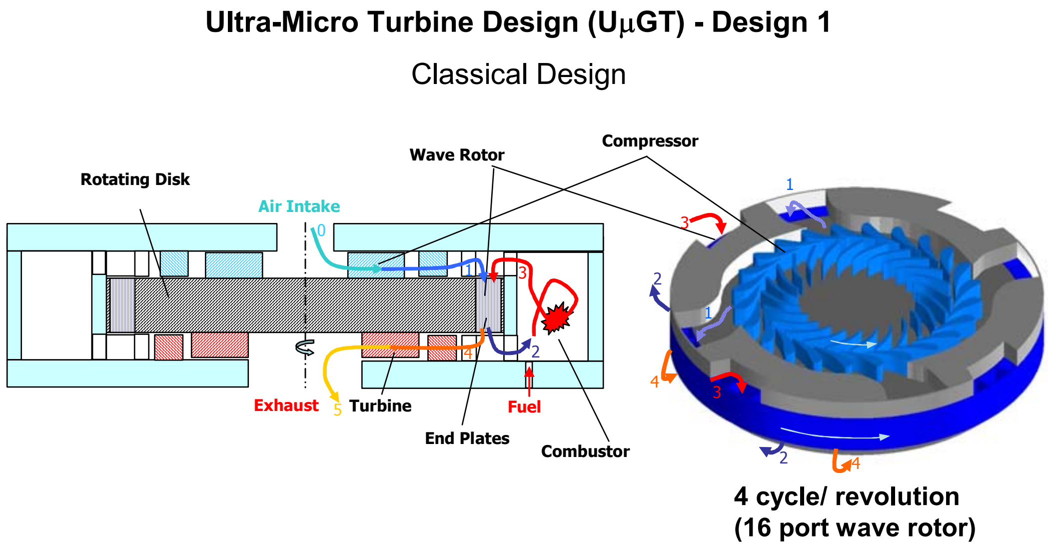

Figure 22: Conceptual design of a ultra-micro gas turbine incorporating a four-port wave rotor, taken from Ref. [224]

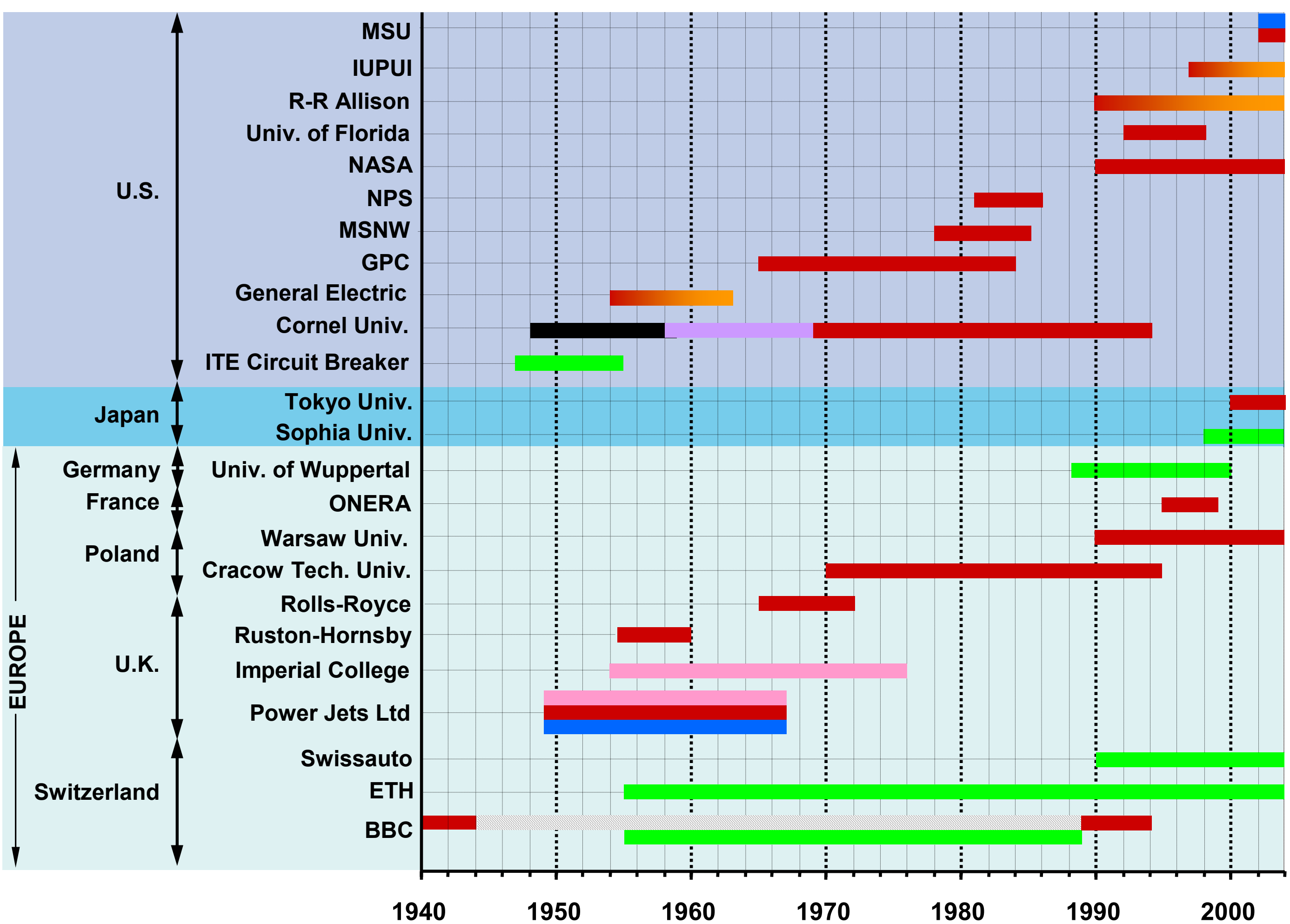

Figure 23: Historical perspective of wave rotor technology. Red: gas turbine application, Green: IC engine supercharging, Blue: refrigeration cycle, Pink: pressure divider and equalizer, Purple: wave superheater, Orange: internal combustion wave rotors, Black: general applications

SUMMARY

Figure 23 summarizes the history of the wave rotor research reviewed here. The goal of this review was to report the continued interest in wave rotor technology and its wide variety of applications. Some of the latest efforts were discussed in more detail, inspiring further research and development on this topic

Now wave rotors have been investigated again extensively. While this technology has become a state-of-the-art knowledge among the involved researchers, it has not reached wider audiences. New knowledge and technology innovations have provided a new opportunity to consider wave rotor concept as innovative technology for today's gas turbine industry to gain significant performance improvements. Still sealing and thermal expansion appear to be dominant problems despite continued research. Special technology developments and additional research on wave rotors overcoming some of present challenges and limitations are needed.

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