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GEOMETRY ANALYSIS AND EFFECT OF TURBULENCE MODEL ON THE RADIAL ROTOR TURBO-EXPANDER DESIGN FOR SMALL ORGANIC RANKINE CYCLE SYSTEM

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Abstract

Organic Rankine cycle (ORC) is one of the most promising technology for small electric power generations. The geometry analysis and the effect of turbulence model on the radial turbo-expanders design for small ORC power generation systems were discussed in this paper. The rotor blades and performance were calculated using several working fluids such as R134a, R143a, R245fa, n-Pentane, and R123. Subsequently, a numerical study was carried out in the fluid flow area with R134a and R123 as the working fluids. Analysis were performed using computational fluid dynamics (CFD) ANSYS Multiphysics on two real gas models, with the k-epsilon and shear stress transport (SST) turbulence models. The result shows the distribution of Mach number, pressure, velocity, and temperature along the rotor blade of the radial turbo-expanders and estimation of performance at various operating conditions. The operating conditions are as follow: 250,000 grid mesh flow area, real gas model SST at steady state condition, 0.4 kg/s of mass flow rate, 15,000 rpm rotor speed, 5 bar inlet pressure, and 373 K inlet temperature. By using those conditions, CFD analysis shows that the turbo-expander able to produce 6.7 kW and 5.5 kW of power when using R134a and R123, respectively.

Keywords: radial turbo-expander; CFD; k-epsilon; shear stress transport; organic Rankine cycle.

I. INTRODUCTION

Climate change concerns coupled with high oil prices are driving research and development on renewable energies such as solar and geothermal energy. One of electric power generation technology that can be used to utilize renewable energy potency in remote area is organic Rankine cycle (ORC). ORC uses organic fluid as the working fluid to provide higher thermal cycle efficiency compared to the conventional steam Rankine cycle at temperature heat source below 300°C. ORC has been studied as the utilization of waste heat recovery [1, 2], solar energy [3], the combination of heat and power (CHP) [4], geothermal [5], and heat recovery from the exhaust gases from the engine [6]. The results of experimental studies show that the small-scale units ORC has a promising performance for power generation especially in remote areas [7].

The ORC could provide a wide output power range, but consist of less components, so that the ORC could be more compact and smaller in size compared to conventional electric power plants. Most studies were focused on the thermodynamic analysis of the ORC, and the selection of the working fluid, with particular attention to obtain the best efficiency of power generation. On the other hand, only few published papers discuss on the design and geometry optimization of turbo-expander. Basically, for output power ranges from 5 to 5,000 kW two types of turbines are proposed, axial or radial turbines. The latter turbine type is considered more attractive, since it has better performance at small output capacity. In basic analysis for optimizing ORC system, isentropic efficiency of expander is specified as a constant value for all working fluids and all conditions. The constant value for all working fluids makes error in theoretical analysis of ORC system. By using different working fluids, the expander has optimum value and different

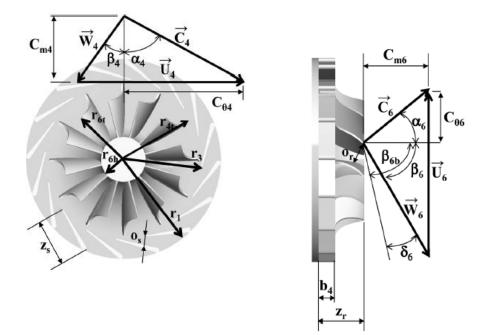* Corresponding Author. Tel: +62-22-2503055
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Figure 1. Geometry parameters of radial turbo-expander

performance [8]. Therefore, it is important to study internal analysis of turbo-expander for developing knowledge on geometry design and its relation to the performance of radial turbo-expanders in ORC systems. The objectives of this study are to calculate the geometry of a radial inflow turbo-expander for small ORC system and predict its performance for various working fluids and operating conditions.

II. GEOMETRY ANALYSIS OF RADIAL ROTOR TURBO-EXPANDER

Baines's method [9] is normally used for preliminary geometry design of radial turbines. This paper deals with a radial turbo-expander shown in Figure 1, where important parameters for geometry design can be observed [10]. Load coefficient ψ and flow coefficient ϕ can be calculated based on the velocity U_4 and can be calculated using Euler's turbo machinery equation as in equation (1).

$$\psi = \frac{\Delta h_0}{U_4^2} = \frac{C_{\theta4}}{U_4} - \varepsilon \frac{C_{\theta6}}{U_4} \qquad (1)$$

where ψ is load coefficient, Δh_0 is entalphy drop in isentropic expansion or specific work output (J/kg), U_4 is blade speed at inlet condition (m/s), $C_{\theta4}$ is absolute tangential velocity at inlet condition (m/s), ε is radius ratio of rotor (r_6/r_4), $C_{\theta6}$ is absolute tangential velocity at outlet condition (m/s).

Figure 2 shows the meridional plane geometry on the radial turbo-expanders. In this model the rotor shroud contour is assumed to be a circle, while the rotor hub to the contours is described as

elliptical geometry assumptions made by Glassman [11]. By assuming that magnitude of outlet swirl is very small, the load coefficient can be predicted as in equation (2). Hence the rotor inlet velocity triangle at station four (inlet) is defined in Equation (3) to (5).

$$\psi \cong \frac{C_{\theta4}}{U_4} \qquad (2)$$

$$C_{m4} = \xi\phi U_4 \qquad (3)$$

$$C_4 = \left(C_{m4}^2 + C_{\theta4}^2\right)^{\frac{1}{2}} \qquad (4)$$

$$\alpha_4 = \tan^{-1}\left(\frac{C_{\theta4}}{C_{m4}}\right) \qquad (5)$$

where C_{m4} is absolute meridional velocity at inlet condition (m/s), ξ is meridional velocity ratio, ϕ is flow coefficient, C_4 is absolute velocity at inlet condition (m/s), α_4 is absolute flow angle at inlet condition (degree).

The static temperature T_4 and the static pressure P_4 at the inlet to the rotor are calculated from Equation (6)-(10).

$$\beta_4 = \tan^{-1}\left[\frac{(C_{\theta4} - U_4)}{C_{m4}}\right] \qquad (6)$$

$$T_4 = T_{04} - \frac{C_4^2}{2C_p} \qquad (7)$$

$$p_4 = p_{04}\left(\frac{T_4}{T_{04}}\right)^{k/(k-1)} \qquad (8)$$

$$r_4 = \frac{U_4}{\omega} \tag{9}$$

$$b_4 = \frac{\dot{m}}{\left[2\pi r_4 \rho_4 C_{m4}\right]} \tag{10}$$

where β_4 is relative flow angle at inlet condition (degree), T_4 is static temperature at inlet condition (K), T_{04} is total temperature at inlet condition (K), C_p is specific heat at constant pressure (J/kg.K), P_4 is static pressure at inlet condition (bar), P_{04} is total pressure at inlet condition (bar), k is ratio of specific heats, r_4 is radius rotor at inlet condition (m), ω is angular velocity (rpm), b_4 is blade height at inlet condition (m), $\dot{m}$ is mass flow (kg/s), ρ is density (kg/m^3). $T_{04} = T_{01}$ (total temperature at inlet, K) and $P_{04} = P_{01} - \Delta p_0$ (total pressure at inlet, bar) Δp_0 is the total pressure loss in the stator (bar).

Inlet area A_4 is given by equation (11). The number of blades Zr and total to static isentropic efficiency η_{ts} are defined as shown in equation (12)-(13) [11].

$$A_4 = \frac{\dot{m}RT_4}{p_4 C_{m4}} \tag{11}$$

$$Zr = \frac{\pi(110 - \alpha_4)\tan(\alpha_4)}{30} \tag{12}$$

$$\eta_{ts} = \frac{\Delta h_0}{\Delta h_0 + \Sigma h_{loss}} \tag{13}$$

where A_4 is area at inlet condition (m), R is universal gas constant (8.314 kJ/kmol.K), Zr is blade number, $\sum h_{loss}$ is total enthalpy losses (J/kg), η_{ts} is total to static efficiency. Results of geometry design are shown in Table 1. Figure 3 shows model for radial rotor turbo-expander with geometry obtained from the design process.

III. CFD SIMULATION

CFD simulation was done with various number of grids to know the influence of number of grid on the simulation speed and accuracy. Table 2 shows operational condition of the radial turbo-expander the grid number used in the simulation.

ANSYS Multiphysics provide 2 types of turbulence models for real gas, the k-epsilon, and shear stress transport (SST). The influence of these model on the simulation results were investigated. In this paper, simulations were performed at steady state condition using various rotor speeds 15,000 rpm, 20,000 rpm, and 30,000 rpm. After the simulation pressure, temperature distribution and torque variation in the rotor were analyzed, using equation (14) to calculate performance of turbo-expander, where P is power output of turbo-expander (kW).
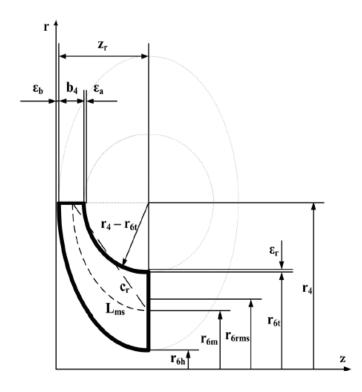
$$P = \dot{m}\Delta h_0 \tag{14}$$

Figure 2. The meridional-plane geometry on the radial turbo-expander

Table 1.
Result for geometry parameter rotor radial turbo-expander

Parameter	Unit	R134a	R123	R245fa	R143a	nPentane
Input Parameter						
Mass flow ($\dot{m}$)	kg/s	0.40	0.40	0.40	0.40	0.40
Angular velocity (ω)	rpm	20,000	20,000	20,000	20,000	20,000
Total temperature at inlet (T_{04})	K	373	373	373	373	373
Number of blades (Zr)	-	12	12	12	12	12
Output Parameter						
Absolute meridional velocity (C_{m4}) (inlet)	m/s	50	44	43	48	49
Blade speed (U_4)	m/s	167	146	143	160	163
Absolute tangential velocity ($C_{\theta 4}$)	m/s	150	132	129	144	147
Absolute flow angle (inlet) ($\alpha 4$)	Degree	71.57°	71.57°	71.57°	71.57°	71.57°
Relative flow angle (inlet) (β_4)	Degree	-18.43°	-18.43°	-18.43°	-18.43°	-18.43°
Absolute velocity (C_4)	m/s	158	139	136	151	155
Relative velocity (W_4)	m/s	52.7	46.38	45.33	50.60	51.60
Inlet temperature (T_4)	K	360.47	361.15	364.37	362.60	367.37
Inlet pressure (P_4)	bar	3.60	3.75	3.93	3.85	4.15
Inlet area (A_4)	m^2	6.5x10^{-4}	4.7x10^{-4}	5.3x10^{-4}	7.7x10^{-4}	8.3x10^{-4}
Radius rotor (r_4)	m	0.100	0.093	0.091	0.101	0.104
Inlet blade height (b_4)	m	0.010	0.008	0.009	0.012	0.012
Inlet density (ρ_4)	kg/m^3	12.238	19.121	17.384	10.737	9.813
Inlet Mach number (M_4)	-	0.87	0.93	0.85	0.75	0.72

IV. RESULT AND DISCUSSION

Numerical simulations using the finite volume method were done to find the fluid flow characteristics, that able to predict the resulted torque, power, and efficiency of the designed turbo-expander. The simulation was done at various of model number of grids, rotational speeds, and working fluids. The number of grid are coarse grid (20,000 mesh), medium grid (100,000 mesh), and fine grid (250,000 mesh). Figure 4 shows velocity contour model with
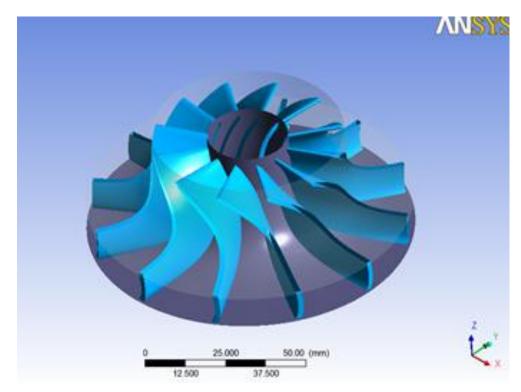

Figure 3. Result geometry radial rotor turbo-expander for small organic Rankine cycle system

Table 2.
Operational condition of the radial turbo-expander and grid number for analysis rotor turbo-expander

Parameter	Value
Inlet temperature (K)	353-423
Mass flow (kg/s)	0.1-1.0
Outlet pressure (atm)	1.0
Outlet temperature (K)	313-373

Table 3.
Power and efficiency for rotor radial turbo-expander with different grid number

Fluids' Parameter	Model 1: 20,000 Mesh	Model 2: 100,000 Mesh	Model 3: 250,000 Mesh
Fluid: R123 (20,000 rpm)			
Torque (Nm)	0.32	0.35	0.36
Power (kW)	6.56	7.08	7.21
Efficiency	0.63	0.64	0.65
Time consuming	6 minutes 30 seconds	11 minutes 56 seconds	28 minutes 5 seconds
Fluid R134a (20,000 rpm)			
Torque (Nm)	0.44	0.47	0.48
Power (kW)	7.94	8.60	8.53
Efficiency	0.63	0.64	0.65
Time consuming	6 minutes 21 seconds	11 minutes 54 seconds	27 minutes 30 seconds

Table 4.
Power and efficiency for rotor radial turbo-expander with different grid number

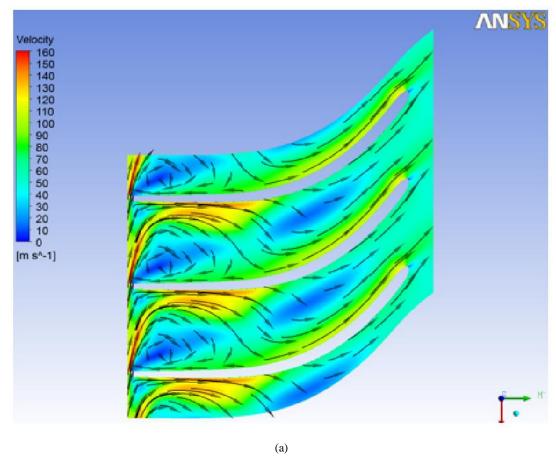| Model	Time consuming (minutes)	Iteration	Power (kW)
k-epsilon (k-ε)	28	200	6.1
SST	28	200	7.2

different number of grids using R123 as working fluid at 20,000 rpm. Model with high number of grid (250,000 mesh), gives better results and shows comparison of the computation time and the number of iterations for the k-epsilon and SST turbulence models. With high number of grid, a strong vortex can be seen. This vortex form a higher pressure area in the region after the leading edge, separating flow from the hub surface and moving it up the blade toward the tip. Other calculation results obtained from the simulation for R123 and R134 working fluids are listed in Table 3.
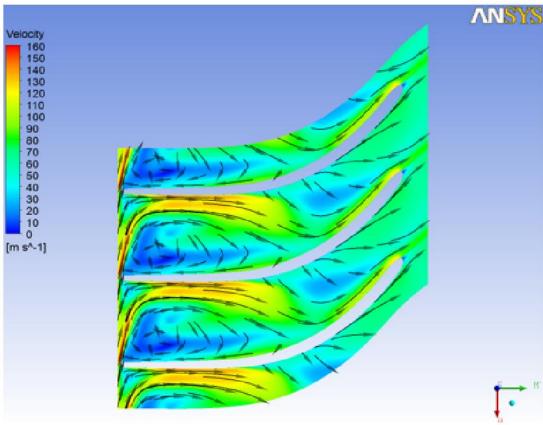
Table 4 shows comparison of the computation time and the number of iterations for the k-epsilon and SST turbulence models. From Figure 4, it can be seen that SST model has good swirl flow prediction on blade passage and results higher power output. For k-epsilon turbulence model, the power ouput is 6.1 kW while for the SST models the power output is 7.2 kW, which is 18% higher. A better model between those two cannot be verified in this study; however experimental results on the expander model may clarify this discrepancy.

The results of an aerodynamic evaluation of rotor radial turbo-expander using SST turbulence model are shown in Figure 5 and Figure 6. This rotor is designed for a low specific speed rotor radial turbo-expander with inlet and outlet ratio under 1. The lower specific speed makes the outlet area of rotor considerably small. This makes the outlet tip of rotor smaller than the inlet area of rotor. The outlet blade's height of rotor is also smaller than the inlet area of rotor [9]. The results are different with conventional characteristic of rotor radial turbo-expander of this type. At rotor with low specific speed using air as working fluid, the fluid flow is more uniform.

Meanwhile, at rotor radial turbo-expander using R123 and R134a as working fluid, a strong vortex is forming on the pressure surface soon after the leading edge, separating flow from the hub surface and moving flow up to the blade toward the tip. It is interesting to note that the pressure gradient introduced in inlet region of the blade passage affect even the flow upstream of the leading edge quite strong. The large variations in flow field that are created in the passage cause very significant changes in the outlet flow (Figure 7). Contours of both relative and absolute flow angle downstream of the trailing edge.

(a)

(b)

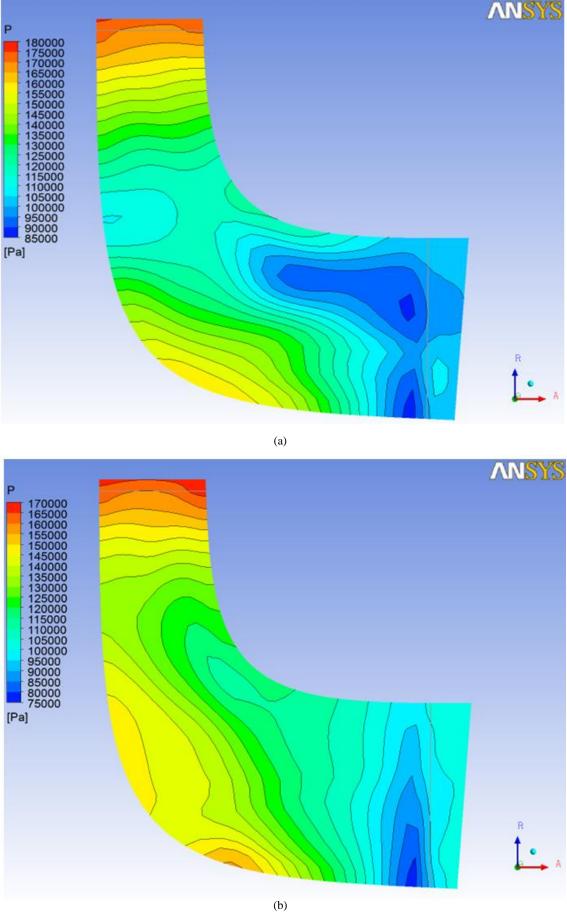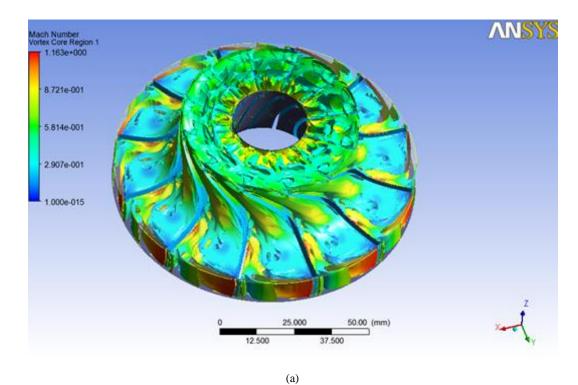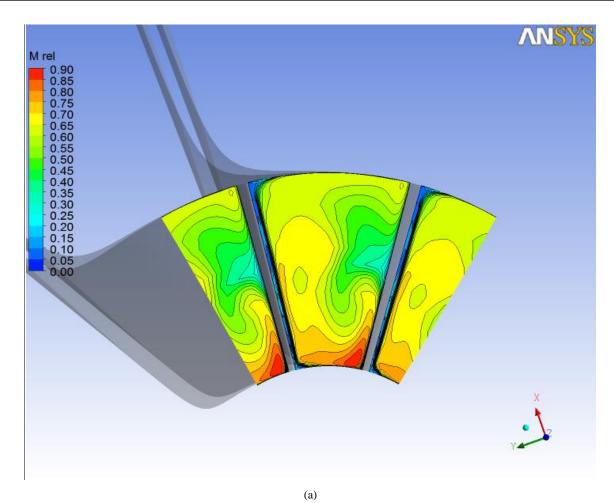Figure 4. Velocity contour for turbulence models: (a) k-epsilon; (b) shear stress transport (SST)
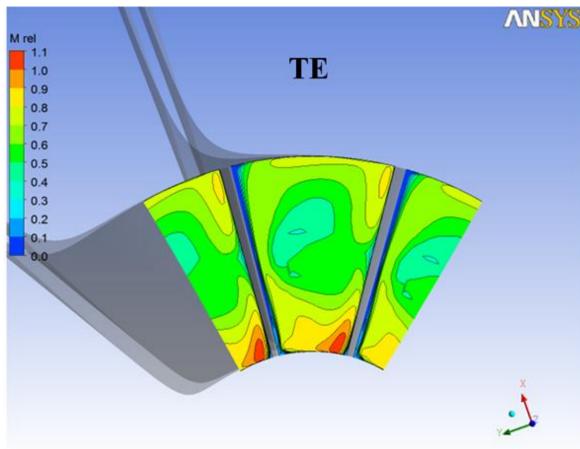
(a)

(b)

Figure 5. Flow prediction at meridional plane for rotor radial turbo-expander at 15,000 rpm using: (a) R123; (b) R134a

(a)

(b)

Figure 6. Flow prediction for rotor radial turbo-expander at 15,000 rpm using: (a) R123; (b) R134a

(a)

(b)

Figure 7. Flow prediction at trailing edge (TE) for rotor radial turbo-expander at 15,000 rpm using: (a) R123; (b) R134a

V. CONCLUSION

This paper present the geometry calculation of a small radial turbo-expander for small organic Rankine cycle system. The performances of the turbo-expander have been evaluated using geometry and 3D numerical analysis method. In geometry analysis, the performance estimation is calculating in iterative loop. The result shows total efficiency-to-static (η_{ts}) for R134a is 0.71 and R123 is 0.66. In simulation using 3D numerical can be concluded that the higher number of grid will give better and accurate results. It is also suggested to use SST models for the numerical analysis to give higher power output. However this results has to be verified in the experimental study. From the two working fluids use in the analysis, R134a gives better performance. At mass flow rate 0.4 kg/s, 15,000 rpm, inlet pressure 5 bar, and inlet temperature 373 K, the expander with R134a produces 6.7 kW power. On the other hands R123 only produces 5.5 kW, respectively.

REFERENCES

[1] T. C. Hung, "Waste heat recovery of organic Rankine cycle using dry fluids," *Energy Conversion and Management,* vol. 42, pp. 539-553, 2001.

[2] Z. Gnutek and A. Bryszewska-Mazurek, "The thermodynamic analysis of multicycle ORC engine," *Energy,* vol. 26, pp. 1075-1082, 2001.

[3] D. Manolakos*, et al.*, "Experimental evaluation of an autonomous low-temperature solar Rankine cycle system for reverse osmosis desalination," *Desalination,* vol. 203, pp. 366-374, 2007.

[4] A. Schuster*, et al.*, "Energetic and economic investigation of Organic Rankine Cycle applications," *Applied Thermal Engineering,* vol. 29, pp. 1809–1817, 2009.

[5] M. Kanoglu, "Exergy analysis of a dual-level binary geothermal power plant," *Geothermics,* vol. 31, pp. 709–724, 2002.

[6] M. Talbi and B. Agnew, "Energy recovery from diesel engine exhaust gases for performance enhancement and air conditioning," *Applied Thermal Engineering,* vol. 22, pp. 693–702, 2002.

[7] M. Imran*, et al.*, "Thermo-economic optimization of Regenerative Organic Rankine Cycle for waste heat recovery applications," *Energy Conversion and Management,* vol. 87, pp. 107–118, 2014.

[8] L. Pan and H. Wang, "Improved analysis of Organic Rankine Cycle based on radial flow turbine," *Applied Thermal Engineering,* vol. 61, pp. 606–615, 2013.

[9] N. C. Baines, "Radial Turbine Design," in *Axial and Radial Turbines*, ed: Concepts Eti, Inc., 2003, pp. 199–327.

[10] C. A. M. Ventura*, et al.*, "Preliminary Design and Performance Estimation of Radial Inflow Turbines: An Automated Approach," *Journal of Fluids Engineering,* vol. 134, pp. [031102-1]-[031102-13], 2012.

[11] A. J. Glassman, *Turbine Design and Application*: NASA, 1994.