THERMODYNAMIC AND OPTIMAL ANALYSIS OF TURBINE-BLEEDING ORGANIC RANKINE CYCLE

[1]KYOUNG HOON KIM, [2]CHUL HO HAN

[1]Department of Mechanical Engineering,
[2]Department of Mechanical System Engineering,
Kumoh National Institute of Technology, Korea
E-mail: chhan@kumoh.ac.kr

Abstract - Organic Rankine cycles (ORCs) with turbine bleeding which is coupled to a direct contact heat exchanger are typically designated as regenerative ORCs and these cycles closely resemble the ORC with regenerator. This paper presents a thermodynamic and optimal performance analysis of turbine-bleeding ORC with and without regeneration for the recovery of low-grade finite thermal energy. The effects of system parameters and working fluid on the system performance were theoretically investigated. Results showed that the performance of basic ORC can be improved by incorporating both turbine bleeding and regeneration and there exists an optimum turbine bleeding pressure for the maximum second-law efficiency. The system performance under the optimal condition is significantly influenced by the turbine inlet pressure, regeneration, and working fluid.

Index Terms - Organic Rankine cycle (ORC), turbine bleeding, optimal condition, second-law efficiency, exergy.

I. INTRODUCTION

Efficient energy conversion systems becomes more and more important due to the worldwide increasing demand for energy in spite of the limited resources of fossil fuel, ozone layer destruction, and electricity price raising, efficient energy conversion systems. Power generation systems such as organic Rankine cycle (ORC), Kalina cycle (KC) and the other ones using low grade heat sources may be solutions for these problems [1-2]. Because of the ORCs simplicity of the technology, availability of components, reliability, and flexibility, they are considered as a practical solution [3-4]. Besides waste heat, geothermal, solar and ocean thermal energy sources is available to provide a clean alternative to fossil fuel combustion in ORCs. Many researches have been done on geothermal energy as a low-grade heat source for ORCs from the perspective of conventional energy and exergy analysis [5-9]. However, there is the difficulty in assessing the additional complexity of the system. While several advanced cycle designs appear promising from a thermodynamic viewpoint, it is not clear that these represent viable economic solutions. And there is a lack of experimental data from open literature [10].

The cycles with organic working fluids are much better than the cycle with water in converting low grade waste heat to useful work and the cycle has the best performance property with saturated vapor at the turbine inlet for the working fluids with non-positive saturation vapor curve slope [11]. The second law efficiency of a geothermal power plant can be significantly increased in comparison to a power generation due to a combined heat and power generation and the most efficient concept is a series circuit with an organic working fluid that shows high critical temperatures like isopentane [12]. All the working fluids have a similar behavior of the efficiency-condenser exit temperature relationship; however the wet fluids with very steep saturated vapor curves in T-s diagram have a better overall performance in energy conversion efficiencies than that of dry fluids [13].

The dry and the isentropic fluids are most preferred working fluid for the ORC and the basic ORC can be modified by incorporating both regeneration and turbine bleeding to improve its thermal efficiency and reduce the irreversibility. Mago et al. [14] reported an analysis of ORC with turbine-bleeding regeneration using dry organic fluids to convert waste energy to power from low-grade heat sources. Desai and Bandyopadhyay [15] studied an ORC with turbine bleeding using 16 different organic fluids as a working medium. The choice of cycle configuration for appropriate integration with the background process depends on the heat rejection profile of the background process. Yekoladio et al. [16] presented a thermodynamic performance analysis of ORC using turbine bleeding for the recovery of geothermal heat sources to maximize power output and improve the first and second law efficiencies. Kim and Park [17] studied the effects of the turbine bleeding pressure and selection of working fluid on the system performance under the conditions of the critical bleeding fraction. A regenerator could be integrated into the cycle to increase the cycle thermal efficiency, especially when drying fluids are used. The requirement of the uses of a regenerator is that the temperature of the vapor leaving the expander be substantially higher than the

condensing temperature [14]. This study presents a thermodynamic performance analysis of a turbine-bleeding ORC with and without regeneration using internal heat exchanger for recovery of low grade heat source. The effects of the regeneration using an internal heat exchanger, working fluid and the important system parameters such as turbine bleeding and turbine inlet pressure on the system performance were intensively investigated. The exergy destruction ratios at each component as well as the second-law efficiency of the system, and the optimal performance for the maximum second-law efficiency were also comparatively investigated.

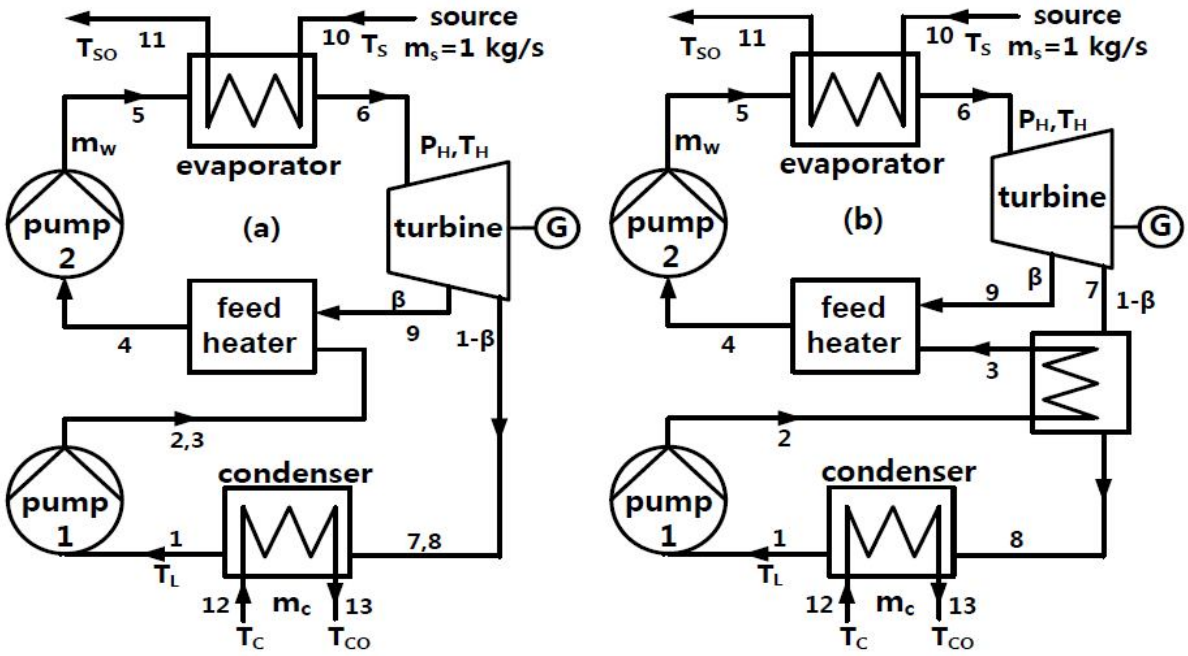


Fig. 1 Schematic diagrams of cycles without (a) and with regeneration (b).

II. SYSTEM ANALYSIS

In this work the thermodynamic and optimal performances of the turbine-bleeding ORC without regeneration (called as basic cycle) and with regeneration (called as regeneration cycle) and the schematic diagrams are shown in Fig. 1 (a) and 1 (b), respectively. For the simulation some assumptions are used as follows [17];

1. The energy source is supplied as a sensible energy at temperature of T_s.
2. The working fluid leaves the condenser as saturated liquid at temperature of T_L.
3. The working fluid leaves the feed heater as saturated liquid.
4. The evaporating temperature is lower than the critical temperature of the fluid.
5. Each of the heat exchangers is operated with a pinch-point condition.
6. Pressure changes besides the turbine and pumps, and heat transfer besides in the heat exchangers are negligible.
7. The isentropic efficiencies of the pumps and turbines are constant.

In the regeneration cycle, the working fluid exits the condenser as a saturated liquid at temperature T_L (state 1). The fluid enters pump 1 and is pressurized to the turbine bleeding pressure P_B (state 2). The fluid is then heated in the regenerator (state 3) while the working fluid out of the turbine (state 7) is cooled (state 8). The fluid is then heated in the feed heater (state 4) by mixing with the working fluid extracted from the turbine (state 9). The fluid is then compressed again in pump 2 to the turbine inlet pressure (state 5). The working fluid is evaporated with the finite heat source and leaves the evaporator as a saturated vapor at temperature T_H (state 6). The source fluid enters the evaporator at T_S (state 10) and comes out at T_{SO} (state 11) after heating the working fluid in the evaporator. The coolant is supplied at temperature T_C (State 12) and exits the condenser at temperature T_{CO} (State 13). For a given mass flow rate of the source fluid m_s, the mass flow rates of the working fluid m_w, and the coolant m_c can be evaluated from the energy balances in the source heat exchanger and condenser. And the bleeding ratio β which is defined as the ratio of the mass flow rate for turbine bleeding to the total mass flow rate of working fluid can be determined from the condition that the fluid leaves the feed heater as a saturated liquid. The heat addition rate Q_{in}, the turbine power production W_t, the pump power W_p, and the net power W_{net}, are determined as follows:

$$Q_{in} = m_s c_s (T_{10} - T_{11}) = m_w (h_6 - h_5) \qquad (1)$$

$$W_t = m_w\beta(h_6 - h_9) + m_w(1-\beta)(h_6 - h_7) \quad (2)$$

$$W_p = m_w(1-\beta)(h_2 - h_1) + m_w(h_5 - h_4) \quad (3)$$

$$W_{net} = W_t - W_p \quad (4)$$

Exergy destruction is the measure of irreversibility of a process and exergy analysis indicates the source of thermodynamic inefficiencies [18]. The exergy is defined as:

$$E = m[h - h_0 - T_0(s - s_0)] \quad (5)$$

Here the subscript 0 represents the dead state. The exergy input to the system by the source fluid, E_{in} = E_{10}, is same as the sum of the net power and the total exergy destruction of the system d_{tot}.

$$E_{in} = W_{net} + d_{tot} \quad (6)$$

The thermal efficiency η_{th}, second-law efficiency η_{II}, and exergy efficiency η_{ex}, are defined as the ratio of the net power to the total energy input, net exergy input and source exergy input, respectively.

$$\eta_{th} = W_{net}/Q_{in} \quad (7)$$

$$\eta_{II} = W_{net}/(E_{10} - E_{11}) \quad (8)$$

$$\eta_{ex} = W_{net}/E_s \quad (9)$$

The parameter D is defined as the ratio of the exergy destruction of a component to the exergy input. The sum of the exergy destruction ratios of the system and the exergy efficiency then becomes unity [12].

$$\eta_{ex} + D_{so} + D_{co} + D_h + D_r + D_c + D_m + D_{tp} = 1 \quad (10)$$

where D_{so}, D_{co}, D_h, D_r, D_c, D_m, and D_{tp} are the exergy destruction ratios of the source exhaust, coolant exhaust, evaporator, regenerator, condenser, feed heater, and turbine/pump, respectively.

Symbol	Component	Expression
D_h	Evaporator	$(E_5+E_{10}-E_6-E_{11})/E_{in}$
D_r	Regenerator	$(E_2+E_7-E_3-E_8)/E_{in}$
D_c	Condenser	$(E_8+E_{12}-E_1-E_{13})/E_{in}$
D_m	Feed heater	$(E_3+E_9-E_4)/E_{in}$
D_{tp}	Turbine/pump	$(E_1+E_6-E_2-E_7-E_9)/E_{in}$
D_{co}	Coolant exhaust	E_{11}/E_{in}
D_{so}	Source exhaust	E_{13}/E_{in}

Table 1 Mathematical expression of the exergy destruction ratios.

In this study four working fluids of isobutane, R245fa, R123, and isopentane were considered and the thermodynamic properties of the working fluids were evaluated by the Patel-Teja equation of state [19-20]. The basic data of the fluids are shown in Table 2, where M, T_{cr}, P_{cr}, and ω are the molecular weight, critical temperature, critical pressure, and acentric factor, respectively [21].

Substance	M (kg/kmol)	T_{cr} (K)	P_{cr} (bar)	ω
isobutane	58.123	408.14	36.48	0.177
R245fa	134.048	427.20	36.40	0.3724
R123	136.467	456.90	36.74	0.282
isopentane	72.150	460.43	33.81	0.228

Table 2 Basic thermodynamic data of working fluids.

III. RESULTS AND DISCUSSIONS

It is assumed that the source fluid is water with a mass flow rate of 1 kg/s. The basic operation data use for the simulation of this study are as follows; source temperature, T_S = 200°C, turbine inlet temperature, T_H = 190°C, reduced turbine inlet pressure, p_H = 0.8, condensation temperature, T_L = 30°C, coolant temperature, T_C = 15°C, pinch temperature difference, ΔT_{pp} = 8°C, isentropic efficiency of pump, η_p = 0.80, isentropic efficiency of turbine, η_t = 0.80, dead state temperature, T_0 = 15°C, and dead state pressure P_0 = 1 atm.

The reduced pressure is defined as the ratio the pressure to the critical pressure of the working fluid. Each of the thermal and second-law efficiencies contains a peak value with respect to the reduced turbine bleeding pressure (RTBP) for the various working fluids. Fig. 2 shows the effects of the reduced turbine inlet pressure (RTIP) on the optimum RTBP for the maximum second-law efficiency. The optimal RTBP increases with increasing RTIP in the basic cycle without regeneration. In the regeneration cycle, the optimal RTBP increases with RTIP when the working fluid is R123 or isopentane. However, the optimal RTBP exhibits a peak with respect to RTIP when the working fluid is isobutane or R245fa. In addition, the optimal RTBP for regeneration cycle is higher than that the basic cycle.

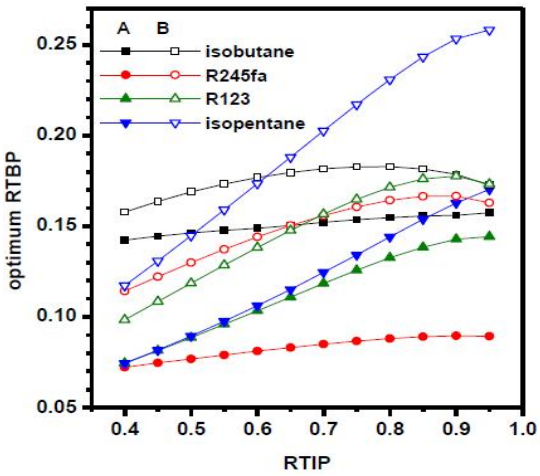


Fig. 2 Effects of RTIP on the optimum RTBP.

Fig. 3 shows the effects of the RTIP on the net power under optimal condition for maximum second-law efficiency. For basic and regeneration cycles, the net power increases with increasing RTIP when the working fluid is isobutane or R245fa; however, it decreases with increasing RTIP when the working fluid is R123 or isopentane. However, it can be seen from the figure that the difference between the net powers of the basic and regeneration cycles is negligible through the whole simulation range of RTIP.

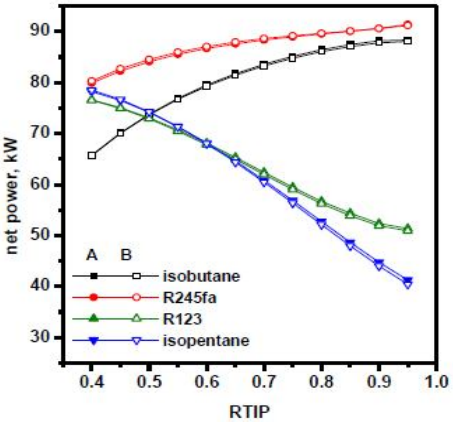


Fig. 3 Effects of RTIP on the net power under optimal conditions.

Fig. 4 shows the effects of RTIP on the maximum second-law efficiency. The maximum efficiency for the regeneration cycle is always higher than the basic cycle. The maximum efficiency may monotonically increase with increasing RTIP or have a peak value with respect to RTIP, depending on the conditions of regeneration, turbine inlet pressure, or working fluid. For example, in both the basic and regeneration cycles, the efficiency increases monotonically with increasing RTIP when the working fluid is isobutane or R245fa; however, it decreases with increasing RTIP when the working fluid is R123 or isopentane. The working fluid showing the highest efficiency is R123 in the basic cycle, while it is isopentane in the regeneration cycle.

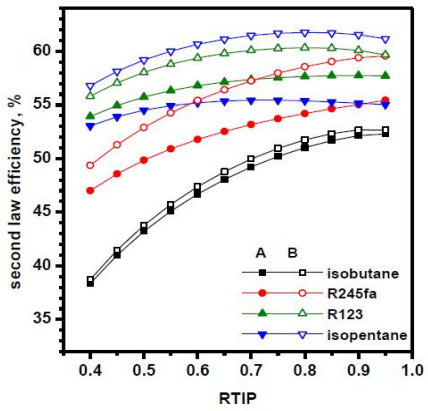


Fig. 4 Effects of RTIP on the second law efficiency under optimal

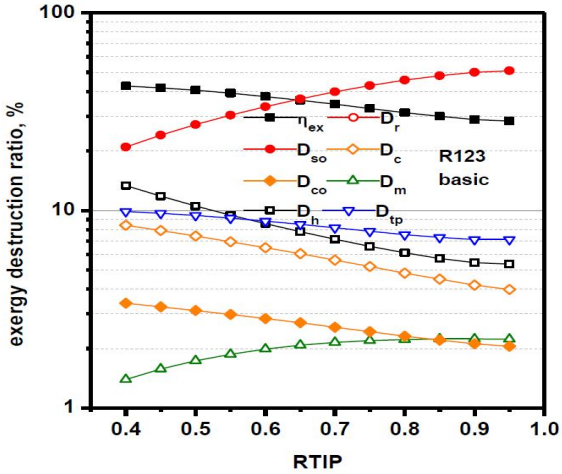


Fig. 5 Effects of RTIP on the exergy destruction ratios in the basic cycle under optimal conditions.

Figs. 5 and 6 show the effects of RTIP on the exergy destruction ratios under the optimal conditions when the working fluid is R123. The exergy destruction ratio of source exhaust (D_{so}) increases with RTIP, and increases from 24.0% at p_H = 0.4 to 53.1% at p_H = 0.95. The exergy destruction ratio of feed heater (D_m) might increase with RTIP or has a peak with respect to RTIP; however, its variation is insignificant due to its small magnitude. The other exergy destruction ratios decrease with increasing RTIP. The greatest exergy destruction ratio besides D_{so} is the exergy destruction ratio of the evaporator (D_h) for low RTIP, and that of the turbine and pump (D_{tp}) for high RTIP.

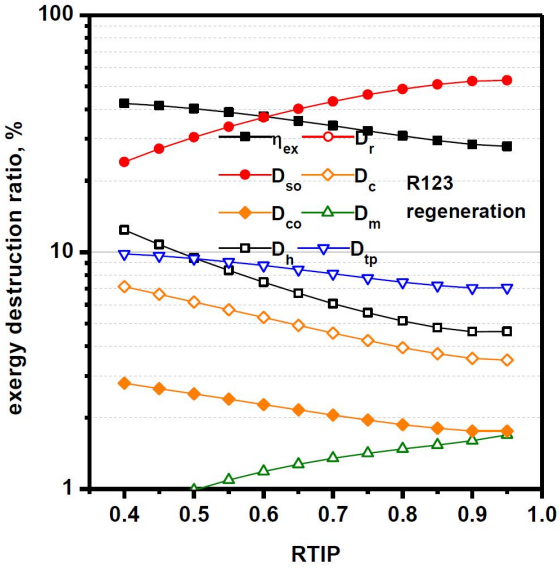


Fig. 6 Effects of RTIP on the exergy destruction ratios in the regeneration cycle under optimal conditions.

CONCLUSIONS

This study presented a thermodynamic and optimal analysis of a turbine-bleeding ORC with and without regeneration for recovery of low-grade heat sources. Water at 200 °C with 1 kg/s was considered as the heat

source, and the four fluids of isobutane, R245fa, R123, and isopentane were considered as the working fluids. The results showed that the second-law efficiency exhibits a peak with respect to the bleeding pressure and the working fluid producing the highest second-law efficiency is R123 in the basic cycle, while isopentane in the regeneration cycle. The exergy destruction ratio of source exhaust is the highest among all of the exergy loss or destruction ratios over the entire range of the simulated conditions, while the second greatest one varies depending on the turbine inlet condition or regeneration.

ACKNOWLEDGMENTS

This paper was supported by Kumoh National Institute of Technology.

REFERENCES

[1] M. Ameri, P. Ahmadi, and A. Hamidi, “Energy, exergy and exergoeconomic analysis of a steam power plant: a case study,” Int. J. Energy Res., vol. 33, pp. 499–512, 2009.

[2] Eurostat, Electricity Prices for Household Consumers, 2013. <http://www.epp.eurostat.ec.europa.eu>.

[3] M. Bianchi and A. De Pascale, “Bottoming cycles for electric energy generation: parametric investigation of available and innovative solutions for the exploitation of low and medium temperature heat sources,” Appl. Energy, vol. 88, pp. 1500–1509, 2011.

[4] H. Nami, A. Nemati, and F. J. Fard, “Conventional and advanced exergy analyses of a geothermal driven dual fluid organic Rankine cycle (ORC),” Appl. Therm. Eng., vol. 122, pp. 59-70, 2017.

[5] B. F. Tchanche, M. Pétrissans, G. Papadakis, G., "Heat resources and organic Rankine cycle machines," Renew. Sustain. Energy Rev., vol. 39, pp. 1185-1199, 2014.

[6] K. H. Kim and H. Perez-Blanco, "Performance analysis of a combined organic Rankine cycle and vapor compression Cycle for power and refrigeration cogeneration," Appl. Therm. Eng., vol. 91, pp. 964-974, 2015.

[7] J. Bao and L. A. Zhao, "Review of working fluid and expander selections for organic Rankine cycle," Renew. Sustain. Energy Rev., vol. 24, pp. 325-342, 2013.

[8] D. Meinel, C. Wieland, and H. Spliethoff, "Effect and comparison of different working fluids on a two-stage organic rankine cycle (ORC) concept," Appl. Therm. Eng., vol. 63, pp. 246-253, 2014.

[9] H. Y. Lee and K. H. Kim, "Energy and exergy analyses of a combined power cycle using the organic Rankine cycle and the cold energy of liquefied natural gas," Entropy, vol. 17, pp. 6412-6432, 2015.

[10] S. Lecompte, H. Huisseune, M. van den Broek, B., Vanslambrouck, and M. De Paepe, "Review of organic Rankine cycle (ORC) architectures for waste heat recovery," Renew. Sustain. Energy Rev., vol. 47, pp. 448-461, 2015.

[11] Y. Dai, J. Wang, and L. Gao, "Parametric optimization and comparative study of organic Rankine cycle (ORC) for low grade waste heat recovery, "Energy Convers. Manag., vol. 50, pp. 576-582, 2009.

[12] F. Heberle and D. Brueggemann, "Exergy based fluid selection for a geothermal organic Rankine cycle for combined heat and power generation," Appl. Therm. Eng., vol. 30, pp. 1326-1332, 2010.

[13] T. C. Hung, S. K. Wang, C. H. Kuo, B. S. Pei, and K. F. Tsai, "A study of organic working fluids on system efficiency of an ORC using low-grade energy sources," Energy, vol. 35, pp. 1403-1411, 2010.

[14] P. J. Mago, L. M. Chamra, K. Srinivasan, and C. Somayaji, "An examination of regenerative organic Rankine cycles using dry fluids," Appl. Therm. Eng., vol. 28, pp. 998-1007, 2008.

[15] N. B. Desai and S. Bandyopadhyay, "Process integration of organic Rankine cycle," Energy, vol. 34, pp. 1674-1686, 2009.

[16] P. J. Yekoladio, T. Bello-Ochende, and J. P. Meyer, "Thermodynamic analysis and performance optimization of organic Rankine cycles for the conversion of low to moderate grade geothermal heat," Int. J. Energy Res., vol. 39, pp. 1256-1271, 2015.

[17] K. H. Kim and S. H. Park, “Thermodynamic performance analysis of organic Rankine cycle with turbine bleeding under condition of critical bleeding fraction,” IJMMME, Vol. 3, pp. 208-212, 2015.

[18] A. Bejan, "Advanced Engineering Thermodynamics," 3rd ed., John Wiley & Sons, New York, NY, USA, 2006.

[19] T. Yang, G. J. Chen, and T. M. Guo, "Extension of the Wong-Sandler mixing rule to the three-parameter Patel-Teja equation of state: Application up to the near-critical region," Chem. Eng. J., vol. 67, pp. 27-36, 1997.

[20] J. Gao, L. D. Li, Z. Y. Zhu, and S. G. Ru, "Vapor-liquid equilibria calculation for asymmetric systems using Patel-Teja equation of state with a new mixing rule," Fluid Phase Equilibria, vol. 224, pp. 213-219, 2004.

[21] C. L. Yaws, "Chemical Properties Handbook," McGraw-Hill, New York, NY, USA, 1999.

